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WAVE ROTOR TECHNOLOGY STATUS AND
RESEARCH PROGRESS REPORT

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November 1982

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## ABSTRACT

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## NOMENCLATURE

| | |
|---|---|
| $a$ | Local sonic velocity |
| $p$ | Local pressure |
| $u$ | Local flow velocity in x-direction |
| $v$ | Local flow velocity in y-direction |
| $P$ | Riemann variable |
| $x$ | Spatial coordinate |
| $y$ | Spatial coordinate |
| $t$ | Time coordinate |
| $C$ | 'Physical' characteristic for 1-D unsteady flow |
| $\Gamma$ | 'State' characteristic for 1-D unsteady flow |
| $\rho$ | Density, $kg/m^3$ |
| $\gamma$ | Ratio of specific heats for air |

Subscripts:

| | |
|---|---|
| $\pm$ | Representing 'families' of characteristics |
| $L$ | Left |
| $R$ | Right |
| ref | Reference state |

# I. INTRODUCTION

In looking for 30% improvement in range for cruise missiles, DARPA has recently considered several alternate concepts; the "wave rotor" is the key component in one of these. "Wave rotors", "wave engines", "wave pressure exchangers", "wave equalizers"--use wave propagation through the fluid trapped in a rotor passage to transfer energy from one fluid to another, or between the fluid and the rotor shaft. The flow into and out of the rotor is steady in time at ports which are at precise locations over parts of the rotor annulus. The rotor passages may be axial or helical, and behave similarly to shock tubes.

Such "unsteady energy exchangers" are more complicated than traditional steady-flow gas turbomachinery and hence have received less attention. They are being reexamined now because they offer a unique advantage--they can be self-cooling, since both hot high pressure gas and cold low pressure gas use the same passages for alternate periods of time in the cycle. Also, the inherently poorer performance of steady flow turbomachinery in small sizes has tended to limit the performance of small gas turbine engines.

In August 1981, Cdr. Don Finch of DARPA asked the Turbopropulsion Laboratory (TPL) for an opinion on the potential of the wave engine concept and positive recommendations were given (Ref. 1). The recommendations were followed by a proposal, at DARPA's request, to begin a research program to develop the analytical tools to assess and to design wave rotor devices (Ref. 2). The Office of Naval Research (Dr. A. D. Wood) agreed to monitor a joint DARPA/ONR program through a modification of the work statement of the ONR-sponsored axial compressor flowfields project (Ref. 3).

The proposal was to form a team consisting of Dr. Atul Mathur as full-time investigator, with part-time participation by Dr. S. Eidelman (NRC Associate at NPS), Professor J. R. Erwin (NAVAIR Visiting Research Professor in Aeronautics), Dr. R. P. Shreeve (Director, TPL) and Professor M. F. Platzer (Department of Aeronautics). The aim was to lay out an experimental and analytical program which would lead to a proper understanding of the potential and problems of wave rotors. The availability at TPL of an expensive wave rotor apparatus formerly used by Klapproth (Ref. 4) at General Electric and contributed by Professor Erwin, and its compatibility with existing power supplies and high response instrumentation, made it possible to propose a basic experimental program at very low cost. Funding received June 6, 1982 from ONR allowed a study program but not an experimental program to be initiated.* The present report documents the

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*Partial support by Naval Air Systems Command and by the NRC Associateship program for different members of the team is acknowledged.

findings made by the TPL team in the study of wave rotors to 31st October 1982.

The work has concentrated on developing an understanding of wave rotors and of the methods needed and constraints involved in their design and analysis. No effort has been given to cycle analysis or to further justifying the study of the technology. Rather, the promise of the potential engine applications is accepted as proven. This was the major conclusion drawn from a workshop meeting on wave rotor technology held at Mathematical Sciences Northwest (MSNW) in Seattle, Washington, 6-10 August 1982 (Ref. 5), which was attended by three members of the TPL team. Furthermore, a concentrated study of the potential application to cruise missile engines is the purpose of the current program at MSNW, under DARPA sponsorship.

Therefore, in the present report, only a brief review of the evidence for useful engine applications is given in the following Section II. In Section III, account is given of the tools and understanding of wave rotor technology that have been developed so far. A staged experiment is proposed which would provide a vehicle to evaluate the utility of the analytical and computational models. Conclusions and recommendations are summarized in Section IV.

## II. POTENTIAL FOR ENGINE APPLICATIONS

In principle, wave rotors offer the potential for increasing gas turbine cycle temperatures without the usual penalty associated with having to cool the turbine structure. It is pertinent to ask, to what extent has the practicability of using wave rotors in engines been demonstrated? What has been the total practical experience to date? We turn first to published reviews.

Two useful reviews of wave rotor devices have been written (Ref. 6, 7). In 1965, Azoury (Ref. 6) reviewed the basic ideas involved in the wave rotor concept including the fundamental (simplified) wave processes involved in the construction of a useful cycle, wave diagrams (including the port opening and closing processes), high and low pressure scavenge processes, and a discussion of the basic 4-port arrangement and effect of adding additional ports. Azoury also reviewed "some industrial applications", including the gas generator (wave-engine), the supercharger, air compressor, refrigerator and the "equalizer/divider". References were given for several experimental programs known to Azoury. The references omitted the work of Kantrowitz (Ref. 8), Von Ohain (Ref. 9), Pearson (Ref. 10), and Klapproth (Ref. 4).

The more recent review by Rose (Ref. 7) acknowledged Azoury's earlier paper, again reviewed basic ideas and updated the review of potential applications. Thermal power cycle applications and applications in chemical processing were dealt with more extensively. Small gas turbines were mentioned only briefly, and no new examples of experience were quoted. Kantrowitz's basic work (Ref. 8) was noted, but those of Von Ohain (Ref. 9), Pearson (Ref. 10) and Klapproth (Ref. 4) were not.

The history of wave rotor engine experience can be summarized as follows:

Brown-Boveri (1941-43). Based on ideas in Seippel's patents, a direct-flow wave rotor was incorporated in a locomotive gas turbine (Ref. 6). Because the cycle did not give the promised overall efficiency it was revised to use a heat-exchanger and the wave rotor was removed. Brown-Boveri has since developed and is marketing the COMPREX supercharger, but has not pursued the direct-flow wave rotor further. It should be noted in passing that the elements which are critical in the design of superchargers for diesel engines, which result in the selection of the reversed-flow arrangement for example, are somewhat different from those which would be present in the design of a self-cooling gas turbine wave rotor.

Von Ohain (April 1946). In Ref. 9 tests of a wave engine designed for 1.4 kg/sec at full RPM are reported to have been made in which

> . . . the maximum pressure ratio attained amounted to 2.4 at a combustion-chamber temperature of 2200°F. The engine

> was able to drive itself; the effective output was practically zero; the scavenging was not sufficient to remove all the hot-air remnants from the cells. . . . tests were limited to 8000 RPM (or 45% design RPM) . . . because of a deficient welding of the cell walls to the rotor.

The historical background given in Supplement A of Ref. 9 is reproduced here in Appendix A.

Kantrowitz (1949). A comparison of experimental measurements and prediction of the behavior of the flow in a wave rotor were reported in Ref. 8. Kantrowitz also initiated an effort at NACA Langley which was continued by Paul W. Huber (Ref. 11). An engine was built and developed through several stages before competing priorities terminated the project. No publication was issued and the apparatus was sent on loan to G.E. Nett power was produced at "very low efficiency" (Ref. 11). The arrangement of the engine, which used a rotor with curved gas passages, and was fed from a compressed air source, is shown in Appendix B.

Pearson (195?-1960). Reported at the August 1982 Seattle workshop and in Ref. 10, Pearson built a wave engine which output 45 horsepower at a cycle efficiency of 9%. This previously unpublished work, carried out as a corporately funded development project by Ruston-Hornby in Great Britain, demonstrated that the prototype wave engine had superior part-speed performance to a contemporary standard gas turbine engine of comparable power level and, further, demonstrated that wave rotors could be made to work at metal temperatures well below the high inlet gas temperature.

The project showed that the level of knowledge necessary to simply make a wave engine work had been easily surpassed.

Klapproth ( -1962). A research and development program was carried out at G.E. to develop a commercial wave engine. The most recent published account is given in Ref. 4. It is noted that the major problems were considered to be mechanical rather than gas dynamic; rotors failed when they were unshrouded and problems were encountered with end seals due to circumferential distortion. They did achieve self-sustaining operation with nett power output before management terminated the project. The termination was the result of a policy to concentrate only on large engines. The future seen for the wave engine was in smaller sizes (Ref. 12).

General Power Corporation [GPC] (1969-1982). Founded in 1969 explicitly to develop wave engines, this is currently the only known on-going commercial effort. Following the patent filed in 1971 (Ref. 13), the project has proceeded with great difficulty. The engine design is conceptually sophisticated and employs very short rotor passages. As for any wave rotor, success depends critically on whether the flow processes have been correctly calculated. Until operation close to design speed can be sustained (for other than structural reasons) the adequacy of the gas dynamic description can not be tested.

In reviewing what has been achieved toward demonstrating the practical use of wave rotors in engines, the work of Pearson stands out clearly from the rest. His is the only work which resulted in a properly documented measurement of output power and cycle efficiency. (It is entirely regrettable that the documentation has only now become available.) Most important perhaps is the fact that the machine worked when it was first switched on, demonstrating that the analytical approach used in the design was accurate enough to get a prototype machine working. Like Von Ohain (Ref. 9), Huber (Ref. 11), Klapproth (Ref. 12) and MSNW (Ref. 14), Pearson found sealing to be the major problem (and in his case, he felt the only problem) standing in the way of achieving the design performance goal. This should be a matter of some concern to GPC, since the shortness of their rotor will serve to accent the leakage problem.

The only other performance figure for a wave rotor device which can be quoted is the "energy exchange efficiency" of 70% achieved by MSNW (Ref. 14) in a "tailored" cold experiment. It is noted that the geometry in this case, very long rotor passages compared to the area of cross section and mean radius to the passage center also very large compared to passage size, were chosen to satisfy the assumptions underlying the analytical approach. None of the engine programs have used parameters in a similar range.

While the potential of wave engines has been demonstrated definitively by Pearson, the total documented practical experience is collectively small. Each attempt has been an experience in a different range of the available parameters. The greatest effort by far, if the Comprex work of Brown-Boveri is included, has been with axial rotor passages. This is certainly the geometry in which the one dimensional wave propagation approach should apply most accurately. Even constant area, staggered (helical) passages present curvature to the relative flow, and a calculational approach which works for large rotors with nearly axial passages may not be good enough for rotors of small diameter with highly staggered passages. The results obtained by Huber using cambered passages would have been of interest for this reason. Pearson's passages were staggered but at a modest angle (~30° to the axis). The final Klapproth rotor had "straight" passages which were staggered at 60° to the axis, which would provide a more severe test of the purely one dimensional approach.

Thus the state-of-the-art is that the wave engine concept works, that a prototype engine similar to Pearson's in design could be made to work now with reasonable risk, but that little to no knowledge is available to say which range of any parameter is likely to give the best results.

# III. WAVE ROTOR TECHNOLOGY

## A. INTRODUCTION

Since, with the exception of the comprex, relatively few wave machines have been built, the technology is not developed to a point where "preliminary design" and "advanced design" procedures have been formalized. On the contrary, each documented example of a wave engine has involved an early arbitrary decision on the selection of rotor passage size and orientation and an iterative construction of a wave cycle that results in a physically realizable system of ports.

Construction of wave diagrams for wave machines is quite involved, requiring considerable time and effort for each one. Complicated unsteady wave phenomena appear and even the simplest mode of operation requires calculation of an array of wave processes. Approaches adopted by various investigators include the method of characteristics and graphical techniques. Such "classical" methods are valuable in obtaining a proper understanding of the technology but are, at the same time, limited in their ability to arrive quickly at a preliminary design configuration.

If, in the future, rational procedures are to be developed for selecting and optimizing designs, it is necessary to have one set of simplified procedures to select the most desirable overall configuration and a more detailed analysis procedure with which to finally optimize. The simplified procedure would be used to select the wave cycle (and it is noted that passage stagger angle can be varied quite independently for a given wave system to give an infinity of different machines), and the more detailed analysis would be required to optimize the port and passage designs.

In the present work, in order to determine what was available and what was needed, the task was adopted of designing a wave rotor to perform as a cold-cold air "pressure equalizer" (Ref. 6). The constraints were to use an existing rotor with a passage stagger angle of 60°, to use an existing air source at 3 atmospheres of pressure, and to pump air from atmospheric to a common exhaust pressure of 1.5 atmospheres. As a result of the exercise, which will not be documented as such, it is possible in the following paragraphs to provide a review of methods available for the design and analysis of wave rotors. The review includes a new technique which was implemented for preliminary design calculations (Section III.B), progress made toward developing more detailed analytical approaches (Section III.C), and the definition of a particular experiment with which to evaluate, in a controlled way, the accuracy of the analytical flow descriptions. The preliminary design technique and the first proposed "turbine mode" experiment are described in a recently submitted paper given in Appendix C.

## B. PRELIMINARY CYCLE CALCULATION TECHNIQUES

### 1. Riemann Problem Solver

Details of a Riemann problem solver code developed at NPS as a fast and efficient method to carry out preliminary wave calculations are given in Appendix C, together with an example of its application to the proposed "turbine mode" experiment wave cycle. The calculational procedure requires only minutes to carry through, with minimal requirements for computer time and storage; e.g. a typical "Riemann Step" calculation requires 1.12 seconds CPU time on an IBM 370-3033AP computer.

### 2. State Diagram Methods

Unsteady flows in shock tubes and pipes have long been treated by "classical" methods wherein the designer has recourse to the construction of "state" diagrams along with the wave diagram. The state diagram is a particularly useful tool for wave rotors in that it provides the designer with some "feel" for the working of the device. Based on a reference state, the state diagram is usually plotted in non-dimensional form, with the ordinate representing $a/a_{ref} = (p/p_{ref})^{\frac{\gamma-1}{2\gamma}}$ and the abscissa representing $u/a_{ref}$. For the case of one-dimensional unsteady flow of an ideal gas, there are two families of physical characteristics, the $C_+$ and $C_-$ characteristics. Corresponding to the physical characteristics there are two families of state characteristics which relate the changes of the flow speed and of the local speed of sound along the physical characteristics. For example, along the $C_+$ physical characteristic, whose path in the physical plane is given by $\frac{dx}{dt} = u + a$, the values of the flow speed and the local speed of sound (or local pressure level) are related by the equation of the corresponding state characteristic, the $\Gamma_+$ characteristic, which is

$$u + \frac{2}{\gamma-1} a = P = \text{constant}.$$

By choosing scales such that a unit on the ordinate of the state plane is $\frac{2}{\gamma-1}$ times a unit on the abscissa, the state characteristics are oriented at ±45° in the state plane. Following a particular gas in a wave cycle from state to state is thus a matter of following the state characteristics in the state plane, along the ±45° lines. The process can be illustrated for the "turbine mode" experiment described in Appendix C.

The turbine mode wave configuration is shown in Fig. 1, which is taken from Appendix C. The cycle is started at the origin of the state plane, shown in Fig. 2, corresponding to state 1 in Fig. 1. The quiescent gas is accelerated and compressed to state 3 by the incoming high pressure air. This change in state is traced out simply on the state plane by moving along the +45° line from the origin to point 3, which is the point of intersection of the state characteristic and the incoming air "boundary curve", drawn for the specified inlet total pressure (absolute) and wave rotor geometry. The wall boundary condition brings the flow to a stop with a corresponding increase in pressure (through the reflected compression wave), to state 5 in Fig. 1. This change in state is seen on the state plane as line 3-5. Assuming, for simplicity, that the secondary waves in the wave diagram (c-e, e-f, f-g) have a negligible influence on the gas, states 4 and 5 then correspond to states 7 and 8. By letting this high pressure gas out to the outlet boundary conditions changes the state from 8 to 10 for the gas that is being tracked. This change is represented in the state plane as line 5-10, which is the state characteristic for the expansion wave. For the cycle to "close", the reflected expansion wave must bring the gas back to its original state. It is clear from the state plane that this will be the case only if the outlet conditions are at the state defined by point 10, which is the point of intersection of line 5-10 and a line from the origin at -45° representing the state characteristic for the reflected expansion wave. Thus, for example, for some other inlet air conditions, the cycle established could be as shown by (1)-(X)-(Y)-(Z)-(1) in Fig. 2, with the stipulation that the outlet conditions are those of point Z. Each of such "closed" cycles, however, give different outlet conditions, and due to the strong interdependence of the parameters, the wheel speed, output (shaft) horsepower, and mass flow rate are different for each cycle (Fig. 3).

It is noted here that the state plane construction described above is good for small pressure ratios when only weak shocks are involved. If two Mach lines of a continuous compression wave intersect in the physical plane, then, as long as the pressure ratio across the resulting shock is small, the straight state characteristics can be used. If, however, the shock involved is strong, the so-called "shock polar" must be used in place of the straight line characteristic.

Construction of state diagrams can become quite elaborate if supercharger or engine applications are involved. These typically involve two different gases (combustion products and fresh air) with differently oriented families of state characteristics for each gas, if both gases are tracked on one state plane. Nevertheless, the "visual feel" that can be obtained from these diagrams for particular applications is difficult, if not impossible, to get from purely numerical results. The present day Brown-Boveri "comprex" has evolved over two decades of research to a high level of sophistication as a diesel engine supercharger. Even though modern numerical codes are used to calculate wave propagation in the machine, state diagrams are used to assess the computed results qualitatively and to monitor the changes which occur from design modifications such as the incorporation of pockets to provide speed/load range capability. The usefulness of the state diagram approach coupled with modern numerical codes is very evident in Ref. 15.

3. Effect of Stagger

Depending on the particular application for which a wave machine is intended, the rotor may have axial passages or "staggered" passages, i.e. canted at some angle with respect to the axis of the rotor. Depending again on the application of the device, "uniflow" or "counterflow" scavenging is used corresponding to the gas (or gases) entering and leaving the rotor from different ends or from the same end. The "turbine mode" experiment described in Appendix C thus has counterflow scavenging.

Uniflow scavenging enables rotor temperatures to be minimized by bringing all parts of the rotor in contact with cold and hot fluid alternately. This is advantageous for engine applications. Supercharging applications do not usually encounter such high temperatures and may incorporate counterflow scavenging. If the rotor passages are staggered, counterflow scavenging makes work extraction simpler due to the 180° reversal that the gas experiences (thereby bringing about a large change in its absolute tangential velocity component). It can be seen then that the larger the stagger angle, the greater is the potential for work extraction for the same wave configuration. The limiting factor here is the correspondingly more complex port and manifold geometries required for highly staggered rotor passages. The rotor to be used in the "turbine mode" experiment has flow passages set at 60° to the axis. For the cycle described in Appendix

C, with a mass flow rate $\dot{m}$ = 0.754 kg/sec, rotor speed $U_{rotor}$ = 146.2 m/sec, and change in tangential velocity $\Delta V_t$ = 237 m/sec, the ideal power that can be extracted is computed as

$$\dot{W} = \dot{m} \cdot U_{rotor} \cdot \Delta V_t$$

$$= \frac{.754 \times 146.2 \times 237}{778} \simeq 33.6 \text{ h.p.}$$

## C. ENGINEERING ASPECTS--ANALYTICAL

### 1. Wave Propagation Process

The wave propagation process is described in some detail in Appendix C. The emphasis there is on the usefulness of the Riemann Problem Solver code for preliminary design of wave rotor cycles. Once such a cycle has been established, however, there is a need for a "generalized" flow solver to track the wave propagation and the flow point by point through the rotor passages. The flow solver should also incorporate real effects such as friction, heat transfer and leakage losses.

Considered as unsteady and one-dimensional, the governing equations for the flow in the passages are hyperbolic and lend themselves readily to solution by the method of characteristics employing finite difference techniques. However, inherent in the wave processes in real devices are discontinuities such as shocks and contact surfaces. The detection of the initiation of imbedded shocks using the method of characteristics is based on locating the point where characteristics of the same family intersect. If the algorithm used is based on following Mach lines of one family and the pathlines, detection of initiation of shocks of the other family is difficult. An inverse marching method makes detection of shocks of either family difficult. Using an "overall" direct marching algorithm based on following Mach lines leads to fairly easy detection of initiation of shocks, but even in this case, the usefulness of the solution may be sharply curtailed if strong shocks are present in the flow. This is due to the severe reduction in solution points necessitated by the coalescence of compression waves to form strong shocks, leading to an increase in the spacing between solution points. The choice of the grid and the algorithm used thus depends strongly on the specific problem and whether information is available beforehand to apply shock-fitting techniques.

The approach adopted in the present work for numerical modelling of wave rotor applications uses the method of Godunov to solve the one- and two-dimensional time dependent equations of gas dynamics in Eulerian form. The formulation is elaborate, but the method is proven to treat any discontinuities in the flow (including contact surfaces) more realistically than, for example, artificial viscosity methods (Ref. 16). The two-dimensional formulation is used for those regions of the flow where such effects are predominant; for example, rotor cell opening and closing. These results can then be incorporated to modify boundary conditions for the overall one-dimensional scheme implemented to calculate the entire cycle. The following section discusses some preliminary results from a 2-D modelling of the transient flow during cell opening.

Appendix D gives results for the partial structure of the wave propagation process in the simple 4-port cycle (Ref. 17) shown in Fig. D1. The flow parameters were computed using a new method and new code being developed currently at McDonald-Douglas (Ref. 18), which has previously demonstrated an unusual capability of following waves and interfaces from points of inception, through multiple interactions, without distortion. As a trial, part I (the high pressure part) of the cycle was computed for a pressure ratio of 2.0. Times $t_1$ and $t_2$ in Appendix B refer to the closure of the high pressure gas inlet port and the closure of the high pressure air outlet port respectively. The shock wave generated by admission of the high pressure gas at time step zero can easily be tracked for successive time steps through the pressure and velocity diagrams, while the motion of the interface between the compressed air and the hot gas can be identified through the entropy plots. The apparent "smearing" of the interface evident in the plots is the result of using first order (spatial) accuracy for simplicity during the development of the code, and can easily be removed.

### 2. Port Opening and Closing

It is suspected that strong 2-dimensional effects are a major cause for losses observed in the performance of wave machines. These effects are a consequence of finite cell widths and the finite times taken by the cells to fully open or fully close to the various "stator" ports. Although there is as yet insufficient evidence in the form of careful experimental measurements that two-dimensional effects are indeed a major cause for observed inefficiencies, it is likely that significant losses should be attributed to them.

The velocity and pressure contour plots presented in Appendix E are preliminary results of tests toward a numerical 2-D modelling of the cell opening and closing process. The code used is an original approach being developed in-house by S. Eidelman* and uses the 2-D Godunov method mentioned earlier. Initial conditions for the test case were taken to be:

$$p_L = 5 \text{ atm}, \quad \rho_L = 6 \text{ kg/m}^3, \quad u_L = v_L = 0$$

$$p_R = 1 \text{ atm}, \quad \rho_R = 1.2 \text{ kg/m}^3, \quad u_R = v_R = 0$$

The two gases are originally on either side of a "solid" wall, contained between horizontal parallel boundaries. At the start of the integration, a diaphragm is removed from the gap above the wall and the wall is moved with a constant velocity

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*NRC Research Associate under the NPS Foundation Research Program.

of 200 m/sec (which is representative of pitch line speeds of wave rotors). The contour plots shown are for seven different time steps (corresponding to different position of the wall or, equivalently, fraction of the cell opened), starting with the cell being open one-third of a cell width. The time periods shown are real time.

3. Ports and Pockets

A challenging aspect of wave machine design is incorporating load and speed range capability for off-design operation. Closely tied with this is the question of whether such an operational range can be obtained with a basic four-port configuration typical of supercharger or engine applications. No rigorous proof is offered, but it may be shown that serious practical difficulties would inhibit the efficient working of a four-port machine which has fixed port geometry, intermediate heat addition and from which useful work extraction is expected. Appendix F outlines some of the difficulties, taking the early Comprex design as the test configuration.

Static "pockets" in the stator walls are possibly the best design tool to use for operational flexibility of wave machines. The present-day Comprex incorporates three types of pockets ("compression", "expansion" and "gas" pockets) to achieve speed and load range capability (Ref. 15). Pearson used recirculatory "loops" in his wave engine to avoid losses inherent in pocket design, in effect adding additional connected ports.

It is noted again that state diagrams are an invaluable tool in visualizing how these elements help in the operation of wave machines. Effort is being made in the present work to understand how best to design and utilize "pockets" and/or recirculatory loops, and how best to evaluate this understanding experimentally.

## D. ENGINEERING ASPECTS--EXPERIMENTAL

### 1. Approach

A first objective in the present program was to design and conduct an experiment using existing hardware which would serve to first assess and subsequently to extend the present level of understanding. The recommendation following the present review is to set up staged experiments, wherein unit processes of a typical wave machine, namely the turbine cycle and the compressor cycle, are first modelled separately. Depending on the degree of success achieved in these tests, a combined cycle experiment (which can not be simply a combination of the two previous cycles) will follow. In the longer term, "hot" tests related to a then identified potential engine application can be considered. The first and simplest of the proposed experiments involves running the wave rotor as an impulse turbine, as is described in Appendix C.

### 2. Apparatus

A schematic of the experiment is shown in Fig. 4. A detailed engineering drawing of the rotor assembly is given in Appendix G. The drawing shows a half-sectional elevation of the assembled wave rotor apparatus with all major components identified. Other partial views help in visualizing the orientation of the passages in the rotor, of which a photograph is shown in Fig. 5. The rotor and assembly hardware, originally used by the General Electric Company (Ref. 4) is in excellent condition. Not shown in the drawing are the inlet and outlet manifolds which must be modified from existing parts. The main constraints in the design of the inlet and outlet manifolds are that they must supply the cycle-dependent flow angles and peripheral openings at inlet and outlet ports and yet must not interfere with other adjacent components. (Additional constraints would be imposed in the case of flight applications, such as the low frontal area required for cruise missile engines.)

It is noted that the bearing arrangement of one cylindrical roller bearing and a set of duplex angular contact ball bearings (assembled back-to-back) is typical of applications where it is necessary to restrict misalignment or shaft deflection, and where close control of axial play is desired. Clearances between the stator or valve plate faces and rotor faces are kept very small to minimize leakage losses. Any misalignment or axial movement could result in rubbing of the staggered labyrinths machined on the rotor and valve plate faces.

Axial clearances can be controlled by shimming on one side of the rotor. This avoids having to disassemble the entire machine in order to set a new clearance. It is expected that the effect of end clearance, or leakage, will be measured in the course of the study.

Both oil and forced air lubrication/cooling systems may be used.

3. Measurement Program

As shown in Fig. 4 , the rotor will be attached to an air dynamometer in order to provide control over the load output on the shaft. The load can be varied over a range of the rotor speed by throttling the dynamometer and controlling the pressure ratio across the wave rotor. The "off-design" behavior of the rotor can be examined in this way, and the available range and efficiency of operation established.

At the same time the wave and flow structure (and therefore wave timing) within the machine will be measured using an instrumented closure plate on the end remote from the inlet and outlet manifolds. By rotating the plate about the axis, probes (pneumatic, fine-wire thermocouple and Kulite semiconductor) can be used to peripherally "survey" the closed end-wall in the wave cycle (Fig. 1). A comparison of the predicted and measured wave structures, pressure and temperature states can be made.

Pressures and temperatures are to be measured at locations within the ports and in the annular space between the outer casing and rotor shroud. Specific attention will be given to the problem of measuring leakage rates during operation.

All data will be acquired and reduced using the Laboratory's HP 1000 computer controlled high speed data acquisition system. It should be possible eventually to display on-line comparisons of computed and measured flow structure using computer graphics. It is entirely possible that computer graphics could provide the key to obtaining a physical understanding of the on- and off-design behavior of wave rotors and of the relative importance of physical and gas dynamic changes.

# IV. CONCLUSIONS AND RECOMMENDATIONS

To this point in time, the wave rotor has been an interesting concept which has seen relatively few commercially successful applications. In most potential applications however, a difficult development program would have been faced to overcome the competition of an existing, or developing simpler steady-state alternate device. Only in the case of the Comprex has a competitive wave-rotor alternate to a conventional turbomachinery unit been successful. In that particular case, as quoted by Rose (Ref. 7), the Comprex has shown a performance which was superior to a competitive standard turbocharger.

In the case of the Comprex, at the outset, there was a potential for overcoming the slow acceleration response which was characteristic of turbochargers. Without this motivation the present Comprex would surely not have been developed.

A much stronger motivation now faces DOD in the area of cruise missiles. The performance of small gas turbine engines falls far short of larger units; and how can you increase the cycle temperature and not have even worse performance because of cooling? While the wave rotor may not be the perfect answer, it offers such a definite promise of advantage that its development in an engine application should be seriously considered.

The present state of knowledge of wave rotors should be seen realistically. With the exception of the Comprex there have been relatively few such devices, ever. Individuals have pursued different concepts, largely in order to simply get them to work. There is very limited documentation available of the wave rotor efforts which have been made, and almost no records of systematically obtained experimental data. On the other hand, the work which has been done, notably by Pearson, has irrefutably established the feasibility of developing a competitive small wave engine.

The most extensive bank of knowledge on wave rotor technology is at Brown-Boveri Co. in Switzerland. From the paucity of design information published on the Comprex after twenty years of development activity, one can conclude that their design tools will remain proprietary to the company. While it may be possible for DOD to contract a design activity to Brown-Boveri in association with a U.S. engine company, it can not depend exclusively on this sole source. Furthermore the engine application has very different constraints and parameter requirements and it would be unwise to be limited to design tools developed specifically for engine turbochargers.

In this situation, overall it is suggested that two closely related programs are needed:

a) a prototype wave engine definition study for the cruise missile application.

b) basic and applied research to understand, explore and advance the technology of wave rotor devices.

The prototype engine definition study would use current knowledge to select the most promising engine type and generate a proposed hardware design. It is probable that a rotor-component development program would follow, and a prototype engine program if that was successful.

The technology research effort, to develop understanding of wave rotor processes, to develop confidence in analytical and computational tools, to identify key problems and explore solutions, is strongly recommended whatever schedule is adopted in the missile engine program. Both analytical and coupled experimental efforts are needed. It is emphasized that documented experimental experience is lacking most of all, and purely analytical approaches can not identify the problems which need to be solved.

The most technologically productive approach would be to initially associate the programs closely, to initially direct the technology effort toward the cruise missile engine, and to, in effect, coordinate the effort (basic and applied) at several centers toward the earliest realization of the cruise missile application. This would concentrate resources toward attaining a practical engine, but would also serve to educate the more academic investigators into the earliest realization of the most difficult general problems associated with the technology.

It is certain that this area of technology has considerable potential, and computational and experimental tools are available today which should make it feasible to understand and therefore harness the unsteady processes on which the technology is based.

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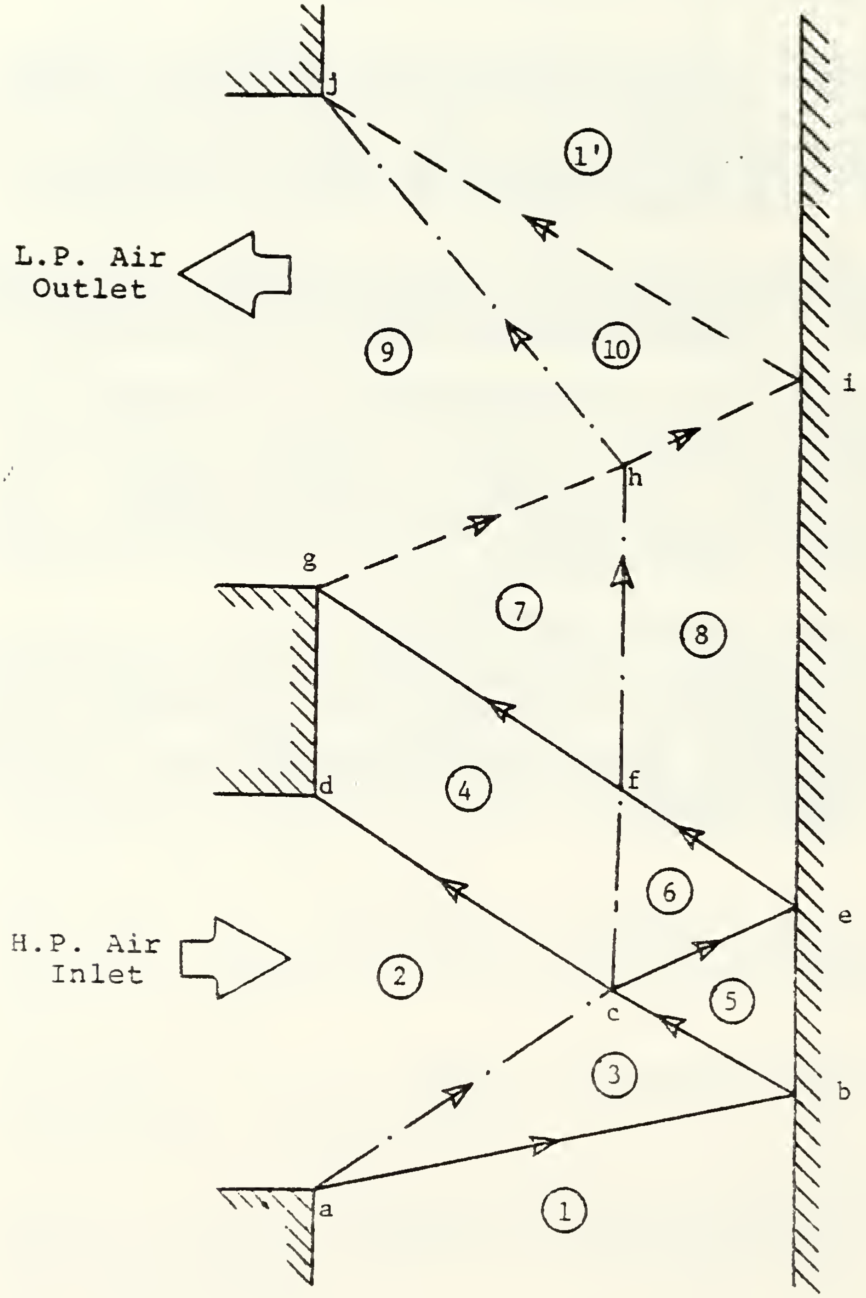



Figure 1. Simplified Wave Diagram for 'Impulse Turbine Mode' Operation

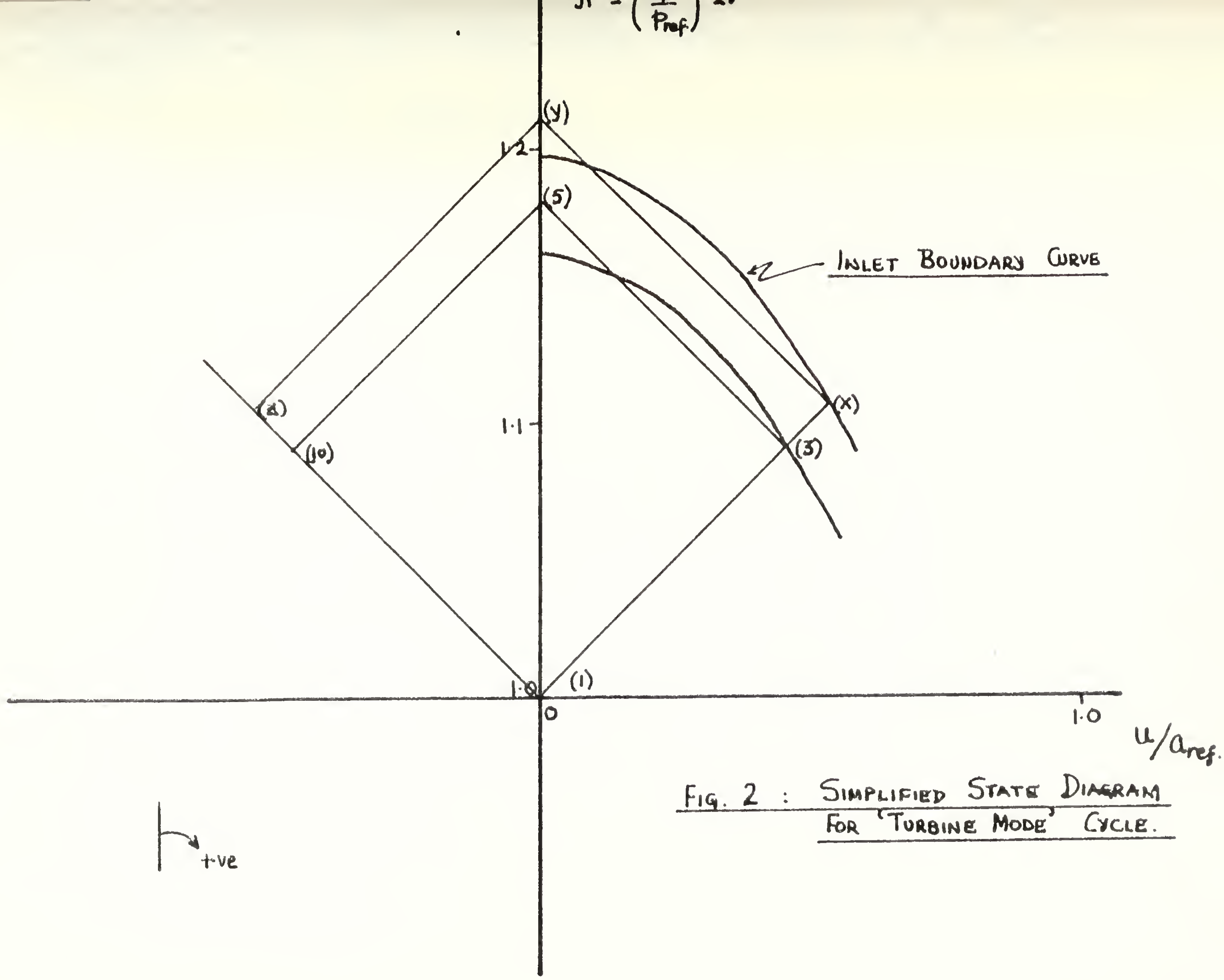



Fig. 2 : Simplified State Diagram
For 'Turbine Mode' Cycle.

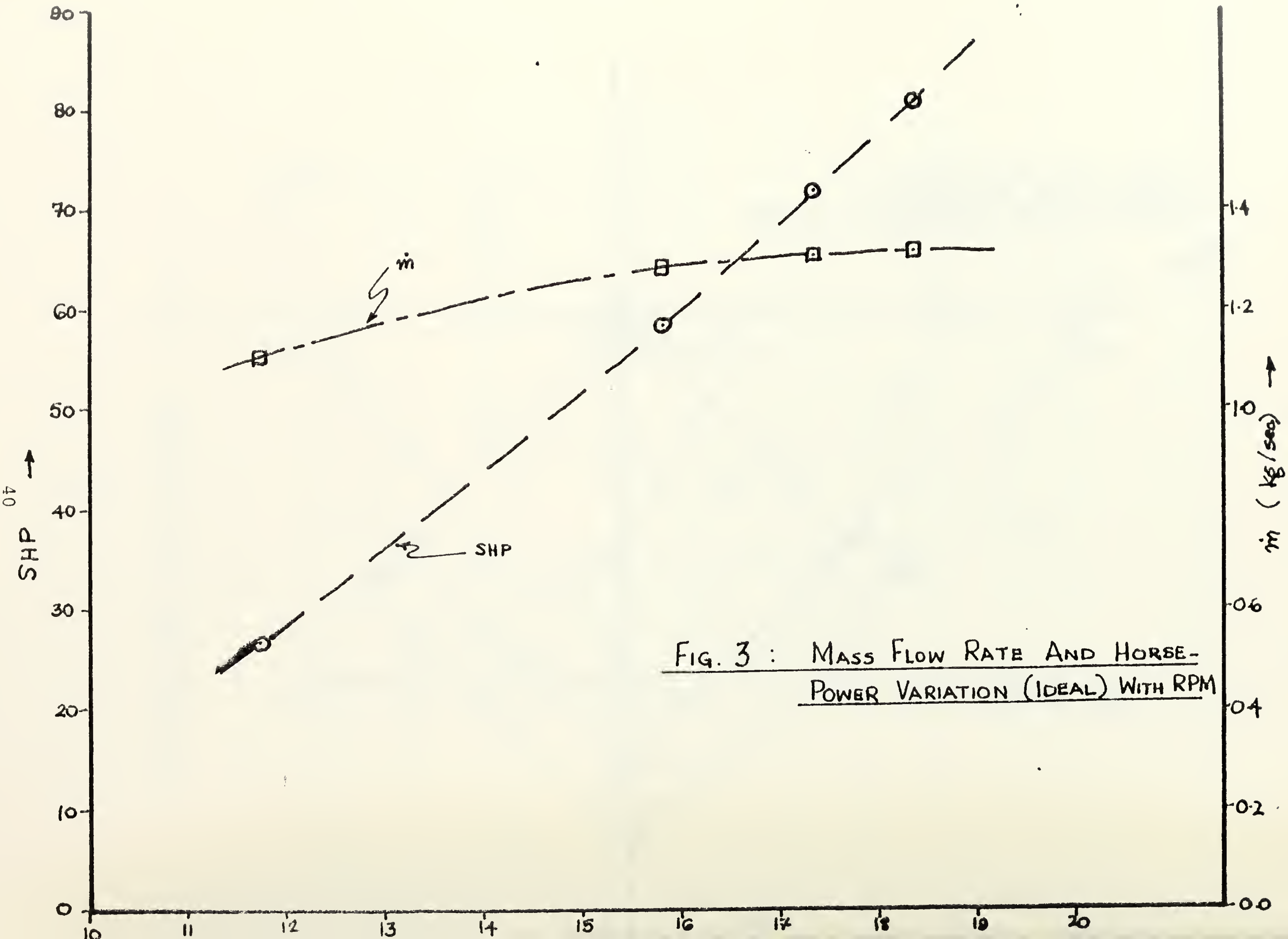



FIG. 3 : MASS FLOW RATE AND HORSE-POWER VARIATION (IDEAL) WITH RPM

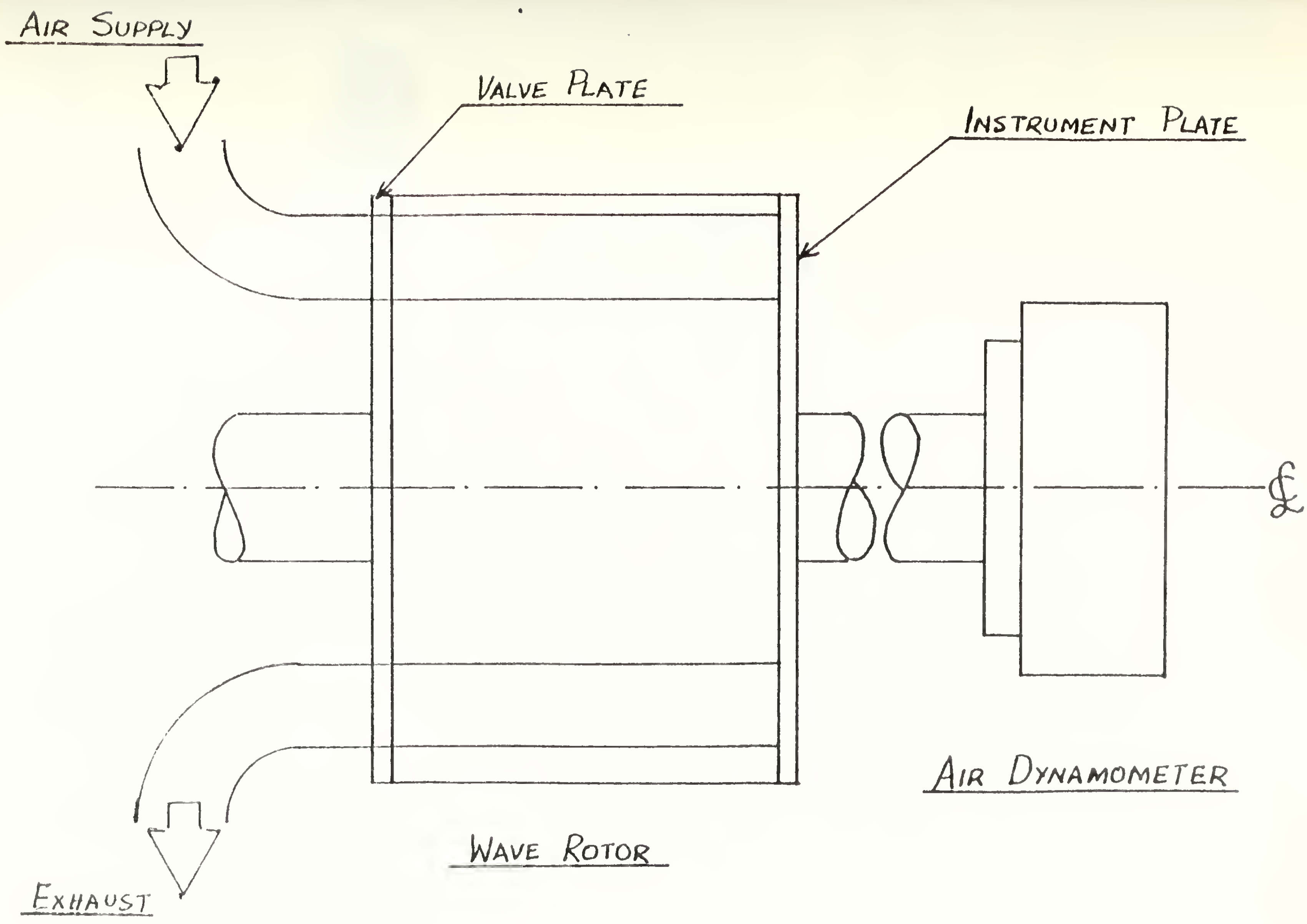



FIG. 4 : SCHEMATIC OF 'TURBINE' EXPT.

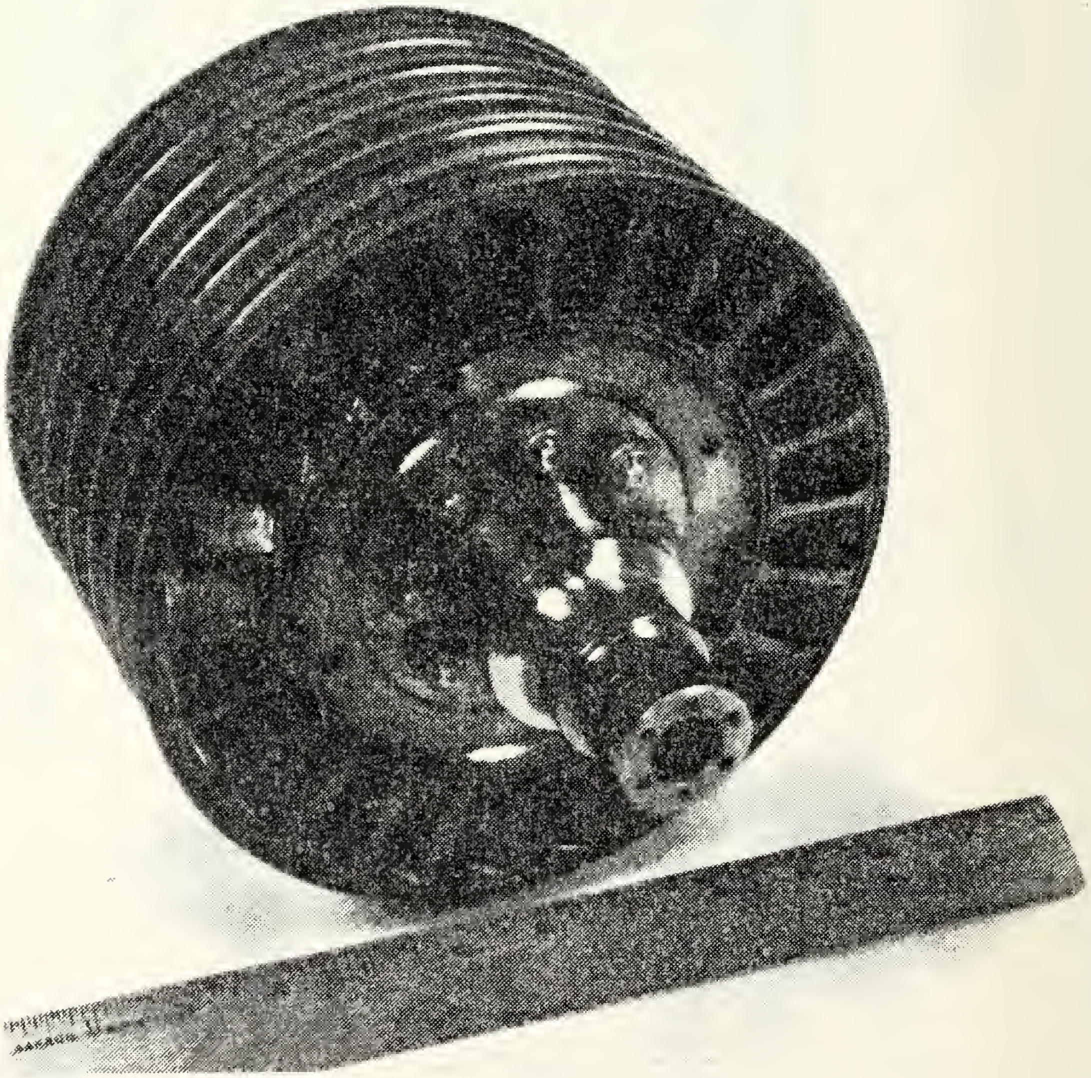


Figure 5. View of the Wave Rotor

## APPENDIX A

## SUPPLEMENT A OF REF. 9

### SUPPLEMENT A

The presented report is supplemented by two further reports translated by the Aeronautical Engine Laboratories, Naval Air Experimental Station, Naval Air Materiel Center Philadelphia:

(a) Report on the Development of a New Heat Energy Principle and Gas Turbine Processes with Pressure Produced by Heating of the Working Medium. AEL Translation 15.

(b) The Heinkel-Hirth RR 2 (Tuttlingen) Gas Turbine Engine, Using Thermal Compression. (Report by Heinkel-Hirth, Stuttgart-Zuffenhausen, December 1946.) AEL Translation 56.

Reports AEL 15 and AEL 56 were not available in their English translation until approximately 4 months after completion of the main portion of F-TR-2186-ND. However, Dr. H. J. Pabst von Ohain consulted German text of these above referenced reports during the preparation of the main report. Several important theoretical problems vital to the thermal compression engine were not treated in these two Navy reports.

The original of AEL 15 was given to the Navy in October 1945, at Stuttgart, Germany. It discloses the general course of development in the field of thermal compression by H. Wolff, along with a brief theoretical description of the principle.

After receiving AEL 15, the U.S. Navy ordered the construction of some test engines by Heinkel-Hirth at Stuttgart-Zuffenhausen. The test results are given in the AEL 56. The theoretical analysis of the processes which are necessary for the evaluation of the test results also are included in AEL 56.

Important theoretical and engineering problems are investigated more completely in the T-2 Report, F-TR-2186-ND, than in the two Navy reports AEL 15 and AEL 56. F-TR-2186-ND is especially concerned with the following:

Unsteady flow processes

Comparison with the pressure exchanger

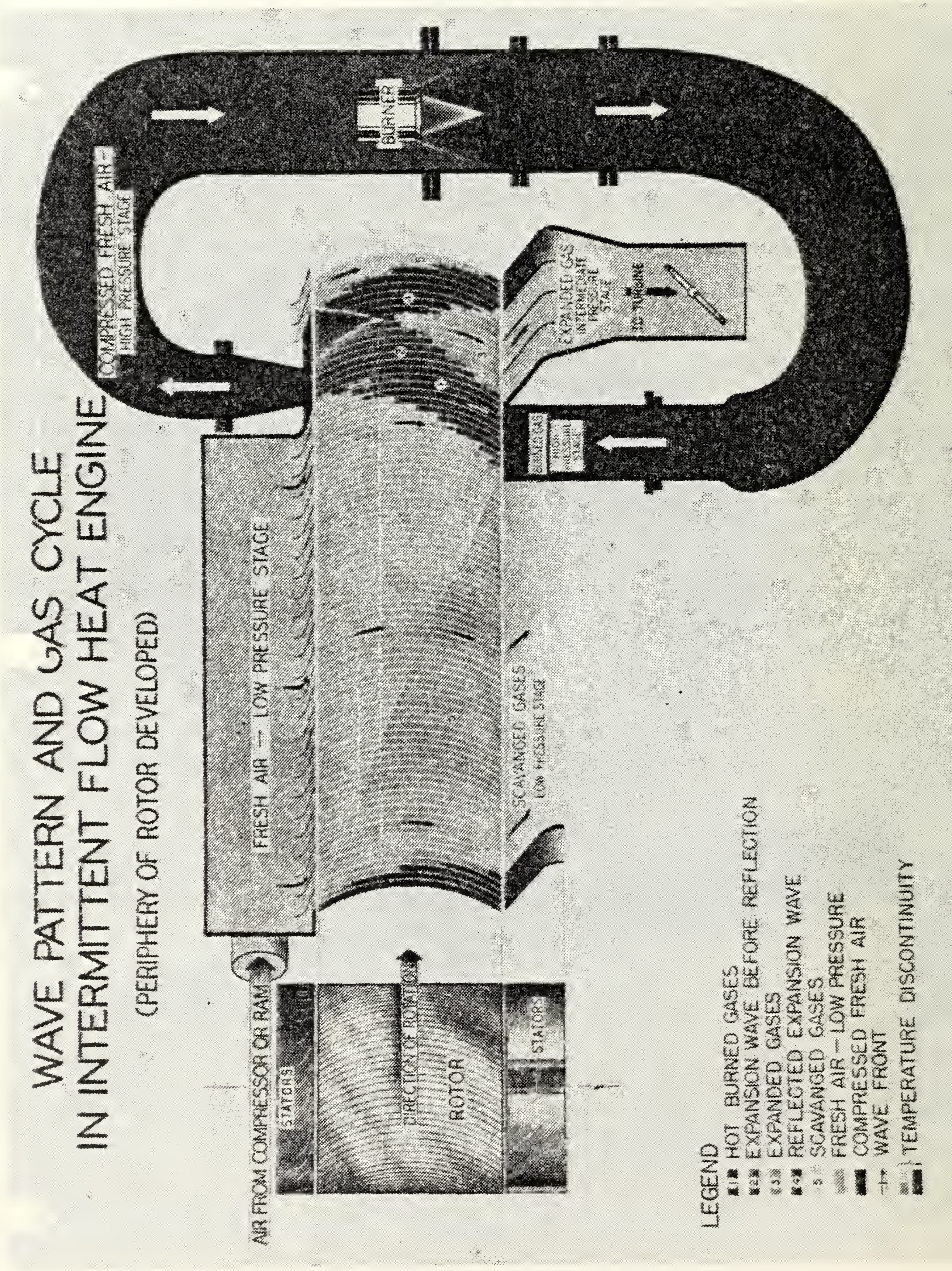
Application for aircraft

Suggestions for new test engines and test program

More detailed description of the method of operation

F-TR-2186-ND, in one volume, analyzes the problems cited above; establishes a technical history to September 1947, and presents a utilization program for the thermal compression engine.

APPENDIX B. ENGINE TESTED BY PAUL W. HUBER AT NACA LANGLEY RESEARCH CENTER, CIRCA 1950



APPENDIX C

# Application of Riemann Problem Solvers to Wave Machine Design§

by

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## NOMENCLATURE

$p$ = pressure, $N/m^2$

$\rho$ = density, $kg/m^3$

$u$ = velocity, m/sec

$x$ = space coordinate (along rotor passage)

$t$ = time coordinate

H.P. = high pressure

L.P. = low pressure

Subscripts:

1, 2, 3, . . . , 10 refer to states shown in Fig. 1

The terms 'tubes', 'passages' and 'cells' have been used interchangeably in the text.

# INTRODUCTION

Wave rotors are devices in which wave propagation is used to effect a transfer of energy between a gas and a rotating shaft or directly between one gas and another. Reviews of such devices and their general operating principles have appeared.(1,2) Commercial applications of such devices have been developed.(3)

In the case of direct transfer of energy, one gas (driver) at high pressure is used to compress a second gas (driven). The process is arranged to occur in tube-like passages arranged on the periphery of a drum, or rotor. The compression is achieved by means of compression waves or shock waves and the compressed gas is drawn off from the end of the tube in which the process takes place. The driver gas then undergoes a series of expansions to a lower pressure and is scavenged out by freshly inducted driven gas at approximately the same pressure level. This fresh 'charge' is then compressed by the high pressure driver gas and the cycle repeats itself.

Steady rotation of the drum sequences the passage of the ends of the tubes past stationary inlet ports, outlet ports and endwalls. This establishes unsteady but repetitive flow processes within the rotating tubes and essentially steady flow in the inlet and outlet ports. The tubes or passages may be oriented axially or at a 'stagger' angle, depending on the application of the particular device. In general, wave machines used as pure pressure exchangers for supercharging purposes have axially oriented passages (e.g. the 'Comprex', Ref. 3) while those with staggered passages may be coupled with conventional turbomachinery or to a drive shaft, since shaft work extraction is possible with this latter configuration.

Complicated unsteady wave phenomena appear in the working of these devices and design of even the most simple mode of operation requires calculation of an array of wave processes such as shock waves, reflected and 'hammer' shock waves (from a wall or moving interface), rarefaction waves (connecting two states of a gas or produced by tube end closure), as well as interactions between incident and reflected waves. In general, the two gases in these devices have considerably different enthalpies, leading to the formation of contact surfaces which also interact with the various waves. The flow in each tube is usually analyzed using one-dimensional or quasi one-dimensional approaches, implying that changes in state occur only along the passage. This enables the wave processes to be depicted on an x-t plane. Such wave diagrams are extremely useful in the examination of possible wave cycles, providing 'visual' information of the wave paths, proper placing of the ports, seeing interactions and whether a particular cycle 'closes': the latter being necessary for cyclic operation. The construction of wave diagrams is usually quite involved, requiring considerable time and effort, since a mismatch in any one parameter gives rise to a new pressure wave which has to be accounted for through the complete rotational cycle. The method of characteristics, despite its limitations (only weak shocks are permitted), has been the technique generally used for carrying out cycle computations.(1) However, graphical methods of constructing wave diagrams are time consuming and may require weeks of tedious calculations.(4) Clearly, a fast and 'user friendly' computational procedure is required to carry out preliminary design calculations for wave rotor devices.

The method described below offers a unified approach to the calculation of wave rotor cycles with no restriction on the strength of the waves involved.

The method has been exercised in the design of a wave-turbine experiment at the Naval Postgraduate School's Turbopropulsion Laboratory. The work is part of a program to examine wave rotors and their potential as components in flight propulsion engines.

## METHOD

The method uses an approach described in Ref. 5 to solve the general 'Riemann problem'--that of finding the flow which results when two gases, each at some specified initial state, are suddenly brought into contact with each other. The initial states are completely defined by specifying the pressure, density and velocity of each gas. Depending on the initial data, four types of discontinuity propagations are possible, involving combinations of shock and and rarefaction waves originating from the point of initial contact and proceeding to the left or to the right. The solution defining the final state is analytical and does not impose restrictions on the initial parameters of the gases. The type of discontinuity which results is defined by the solution and is not prescribed beforehand.

A computer code that solves the Riemann problem for any given initial states of the two gases has been developed. (The program in PASCAL is available from the authors upon request.) The program is set up such that the gas with the higher initial pressure may be on either side. The results give the type of discontinuity, e.g. shock-shock, shock-rarefaction, etc., the pressure and velocity at the interface and the densities on either side of the interface. Velocities of propagation of the waves involved are also computed, with two velocities calculated for rarefaction fans corresponding to the head and tail waves. The pressure at the interface is computed by an iterative procedure, using the average of the two initial pressures as the initial value. Entropy checks are incorporated into the program which disallow the appearance of expansion shocks. Although the program, in its present form, can handle only gases which have the same ratio of specific heats, it can quite readily be modified to deal with different specific heat ratios, as is outlined in Ref. 6. The proposed wave-turbine and subsequent experiments involve only relatively cold air without combustion products, and hence cycle calculations with different specific heat ratios have not been necessary.

The following section describes the wave-turbine as an example of how cycle calculations may be carried out using the program.

## EXAMPLE

Figure 1 shows the wave diagram (not to scale) for the proposed 'turbine mode' experiment in which a wave rotor would be used to work as an impulse turbine and produce shaft power output. For clarity, the waves have been shown as single, straight lines, with full lines indicating compression or shock waves, chain-dotted lines indicating contact discontinuities and dashed lines representing expansion fans. The direction of rotation of the rotor is as indicated by the arrow at the bottom of the figure. The right side of the rotor is blocked off, and the left side tube ends open to inlet and outlet ports alternately. The encircled numbers depict regions of uniform flow but at different state with respect to adjoining regions. Subscripted flow parameters used in the following discussion correspond to these states.

Starting the cycle at state 1 at the bottom of the figure, the rotor tubes are filled with air in a quiescent state and approximately ambient pressure. The cells are then brought into contact with incoming high pressure air at the inlet port. This generates a shock wave (a-b) which propagates into the air at state 1, raising its pressure and density to that of state 3. A mass velocity $u_3$ is also generated behind the shock. An interface (a-c) separates the incoming air and the compressed air, although for this 'cold' configuration the two densities are not very different.

Shock (a-b) reflects off the solid boundary at b as (c-b) and intersects the slower moving interface at point c, where part of the shock is transmitted (c-d) and part is reflected (c-e), creating states 4, 5 and 6. The velocity in state 5 is zero and nearly so in states 4 and 6. The inlet port is closed when the transmitted shock (c-d) arrives at the left end. Shock (c-e) gets reflected again at point e as (e-f) and continues on as (f-g); these 'secondary' reflections are of almost zero strength and do not affect the flow significantly.

At point g, the air outlet port is opened and the compressed air in state 7 exits, with an expansion fan propagating in the opposite direction. The interface bends towards the exit and reaches the end of the passage at point j. The arrival of the interface at point j is timed to match the arrival of the reflected expansion wave (i-j), and the outlet port is closed at this moment. The air in the cells is again in a quiescent state and should be at the same pressure and density as that of the original state 1 for the cycle to be repeated.

The passages in the rotor are 'staggered' at $60^\circ$ to the axis of the rotor, and the change in the angular momentum effected by the reversal in direction of the air flow provides a torque and allows the extraction of shaft work.

The calculations using the Riemann solver code are as follows:

STEP 1: H.P. air at state 2 hitting L.P. air at state 1.

Initial conditions for Riemann problem solver:

$$\boxed{p_2,\ \rho_2,\ u_2 \quad p_1,\ \rho_1,\ u_1 = 0}$$

Discontinuity Type: Rarefaction-Shock

The shock wave propagates to the right into the rotor passage and the rarefaction propagates to the left into the inlet port. A contact surface follows the shock wave into the passage at a slower speed. In the limiting case where no disturbances are to be propagated into the port (for uniform flow conditions), the initial parameters may be varied to obtain the singular condition in which no wave propagates to the left and a shock wave propagates to the right. This can be described as the discontinuity type 'no wave-shock'.

State 3 is then completely defined.

STEP 2: The flow in region 3 is confronted by the 'wall' boundary condition at the right side and is brought to a halt by the reflected shock (b-c). This situation is analogous to the flow colliding with a flow at the same state but with equal and opposite velocity.

Initial Conditions:

$$\boxed{P_3,\ \rho_3,\ u_3 \quad P_3,\ \rho_3,\ -u_3}$$

Discontinuity Type: Shock-Shock

State 5 is completely defined.

STEP 3: The reflected shock (b-c) intersects the interface at point c.

Initial Conditions:

$$\boxed{P_2,\ \rho_2,\ u_2 \quad P_5,\ \rho_5,\ u_5 = 0}$$

Discontinuity Type: Shock-Shock (or Shock-Rarefaction depending on incoming flow temperature)

States 4 and 6 are completely defined.

STEP 4: Reflected shock (c-e) is re-reflected from the wall at point e.

Initial Conditions:

$$\boxed{P_6,\ \rho_6,\ u_6 \quad P_6,\ \rho_6,\ -u_6}$$

Discontinuity Type: Shock-Shock

State 8 is completely defined.

STEP 5: Shock (e-f) hits interface at point f.

Initial Conditions:

$$\boxed{p_6,\ \rho_6,\ u_6 \quad p_6,\ \rho_6,\ -u_6}$$

Discontinuity Type: Shock-Rarefaction

State 7 is completely defined.

STEP 6: Air in the rotor cells is released to exhaust (ambient) conditions.

Initial Conditions:

$$\boxed{p_{amb},\ \rho_{amb},\ u_{amb} = 0 \quad p_7,\ \rho_7,\ u_7 = 0}$$

Discontinuity Type: Rarefaction-Shock

State 9 is completely defined.

STEP 7: Rarefaction wave (g-h) intersects interface at point h.

Initial Conditions:

$$\boxed{p_9,\ \rho_9,\ u_9 \quad p_8,\ \rho_8,\ u_8 = 0}$$

Discontinuity Type: No wave-Rarefaction

States 9 and 10 are defined. State 9 should be the same as the one obtained previously in Step 6.

STEP 8: Rarefaction wave hits the wall at point i, and is reflected in like sense.

Initial Conditions:

| |
|---|
| $p_{10}$, $\rho_{10}$, $-u_{10}$  $p_{10}$, $\rho_{10}$, $u_{10}$ |

Discontinuity Type: Rarefaction-Rarefaction

State 1' is defined. This should match with the original state 1 for 'cycle closure', which is required if the rotor is to operate continuously.

Table I gives the values computed for the cycle and with the procedure described above. The pressures are static values and the velocities are referred to the rotor.

Clearly, a mismatch of rotor speed and/or inlet conditions from design point values will cause new waves to be generated which have to be carried through the entire cycle in order to assess their overall effect on the performance. (This is done in practice to incorporate features such as 'pockets' which make these devices operationally flexible, see Refs. 1 and 3.) For preliminary design purposes, however, having to use a generalized flow solver to arrive at a preliminary wave diagram would be time consuming and expensive. The Riemann program is particularly useful for this purpose because of its 'building block' approach, which allows walking through any wave configuration state by state. Once a viable cycle is established, a detailed flow solver can be applied to incorporate effects of friction, heat transfer and the finite times taken for cell opening and closing.

TABLE I

Computed Values for 'Impulse Turbine' Cycle

| Region | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 1' |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Pressure ($\times 10^{-5}$ $N/m^2$) | .943968 | 1.71453 | 1.71453 | 2.97194 | 2.96339 | 2.97194 | 2.97189 | 2.98056 | 1.71588 | 17.2121 | .94569 |
| Density ($kg/m^3$) | 1.388 | 2.185 | 2.138 | 3.221 | 3.145 | 3.151 | 3.221 | 3.157 | 2.176 | 2.133 | 1.406 |
| Velocity (m/sec) | 0 | 136.8 | 136.8 | 0 | 0 | ≈0 | ≈0 | ≈0 | -136.4 | -137.1 | 0 |

## CONCLUSIONS

The calculational procedure in the example above required only minutes to carry through, with minimal requirements for computer time or storage. (On average, a typical 'Riemann Step' calculation required 1.12 seconds CPU time on an IBM 370-3033AP computer.) The Riemann problem solver code therefore gives a fast, efficient and unified approach to carry out the preliminary design of wave rotor devices with diverse wave structures and pressure ratios.

It is noted that the code may be coded easily on any 'home' computer and does not require external hardware or software libraries.

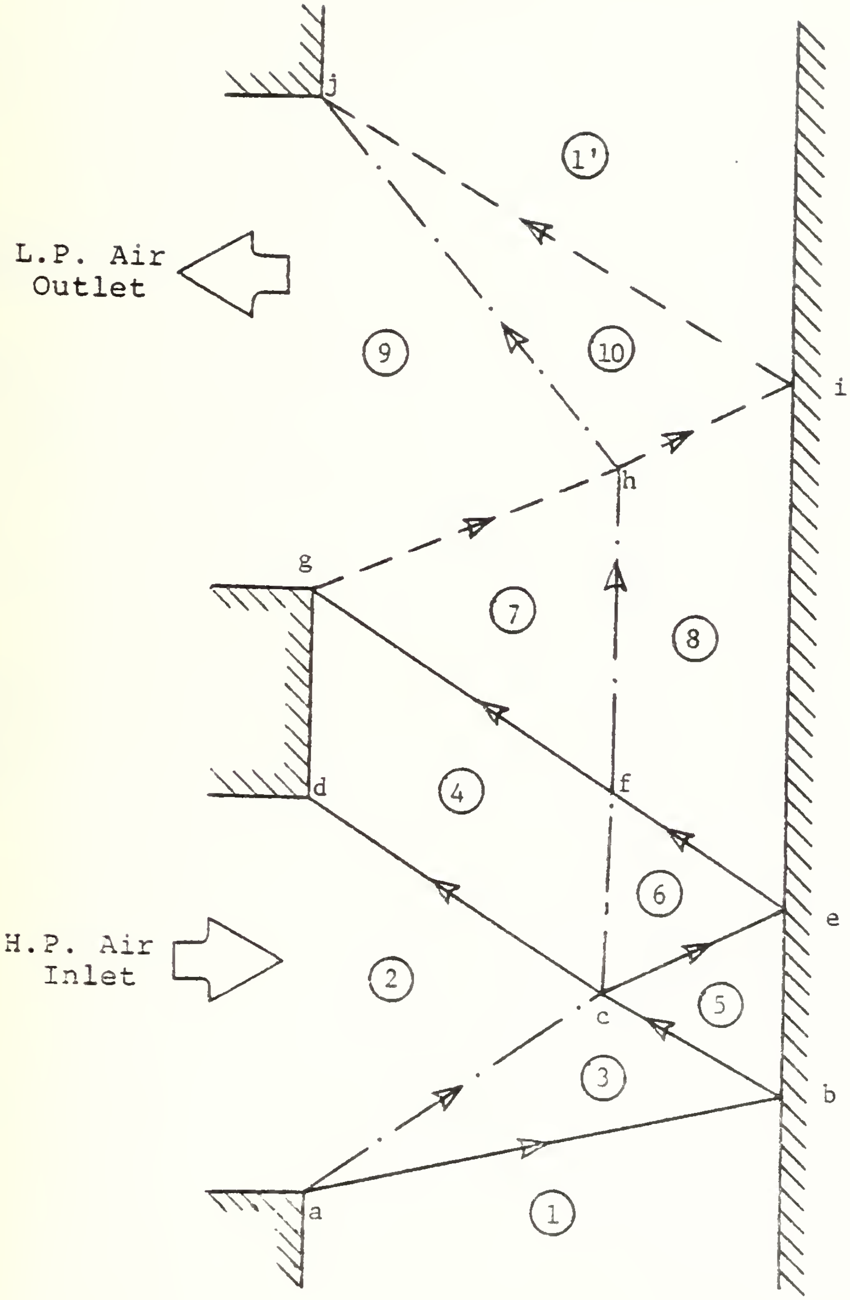


Figure 1. Simplified Wave Diagram for 'Impulse Turbine Mode' Operation

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APPENDIX D

## PRELIMINARY COMPUTATIONAL RESULTS FOR SIMPLE 4-PORT WAVE MACHINE

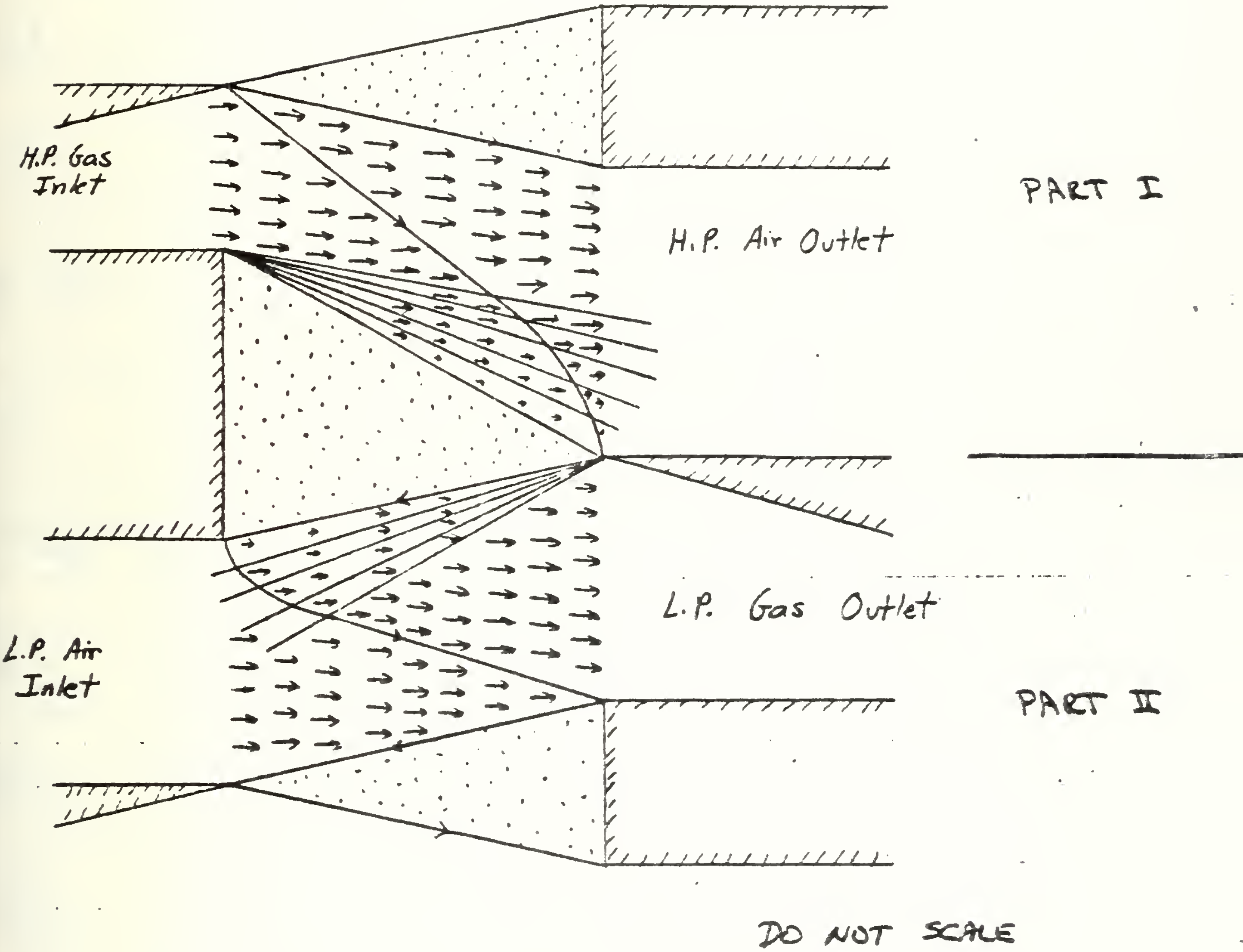



FIG. D1 : WAVE DIAGRAM FOR SIMPLE 4-PORT GEOMETRY

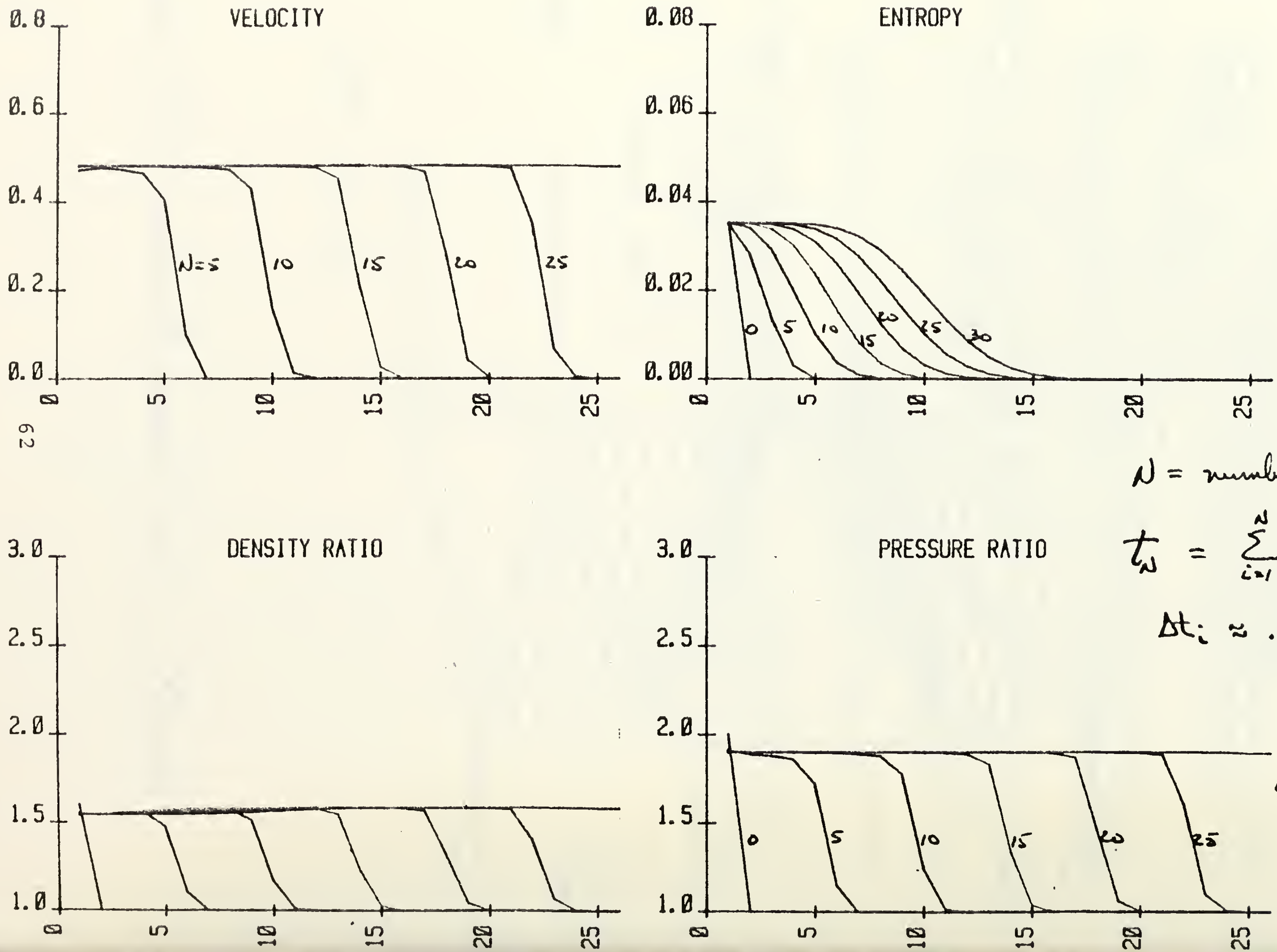
# 1D EULER FORMULATION-WAVE ROTOR

P= 2.00

T= 1.25

t1= 32.00

t2= 56.00




$N$ = number of time steps

$$t_N = \sum_{i=1}^{N} \Delta t_i$$

$\Delta t_i \approx .63$

$N=30$ port opens & shock hits left duct

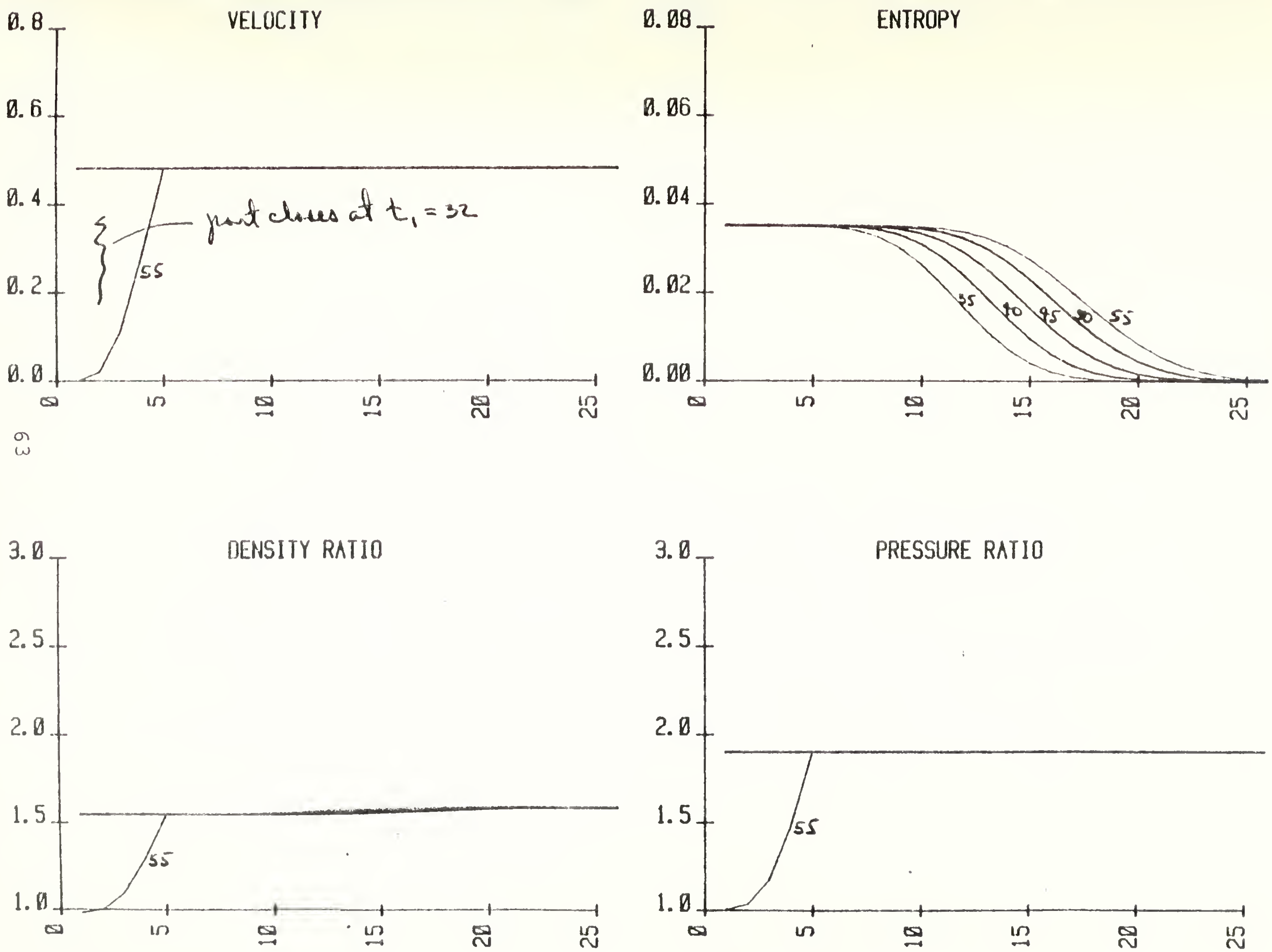
1D EULER FORMULATION WAVE ROTOR

P= 2.00

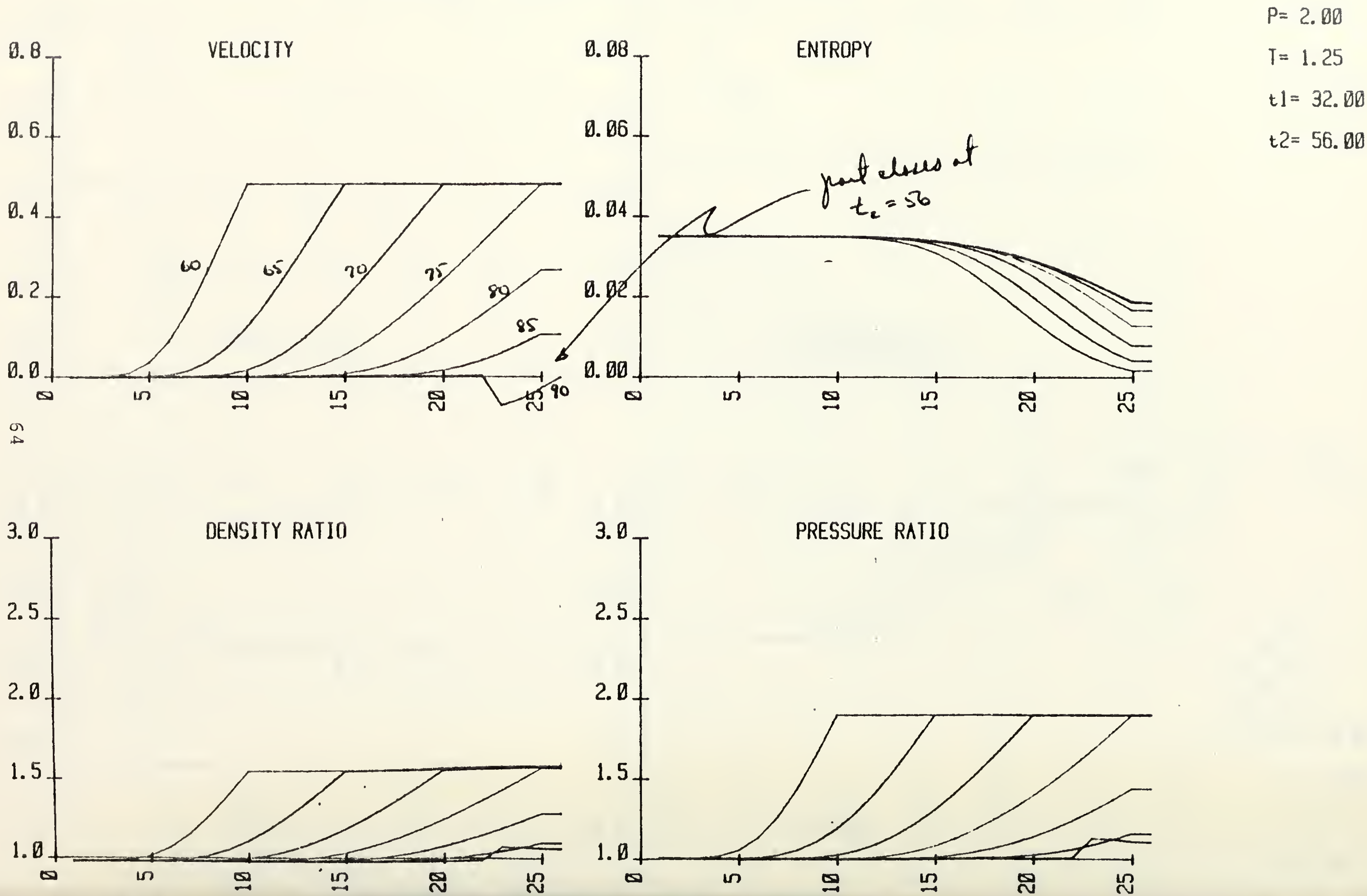
T= 1.25

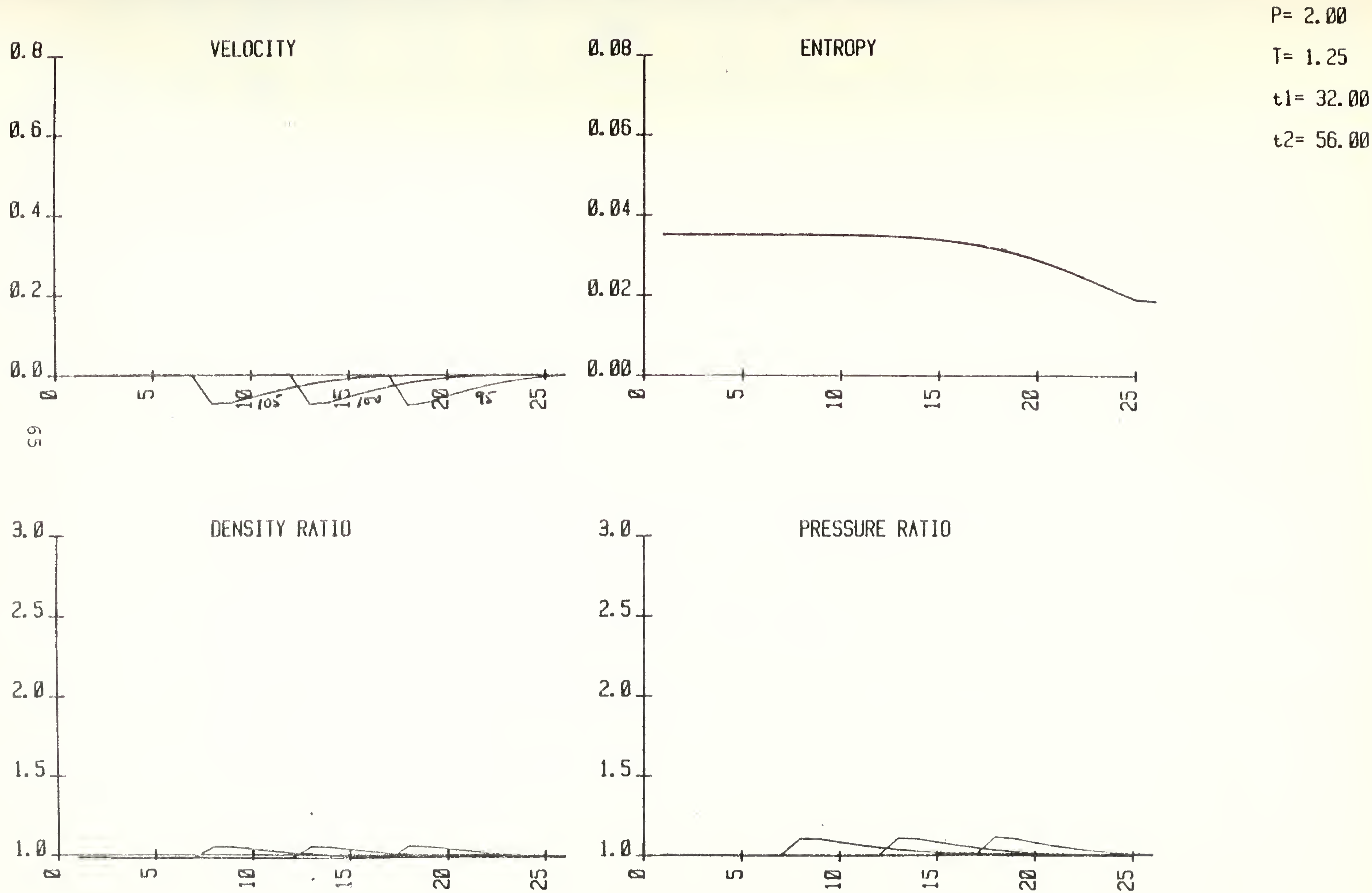
t1= 32.00

t2= 56.00



# 1D EULER FORMULATION-WAVE ROTOR

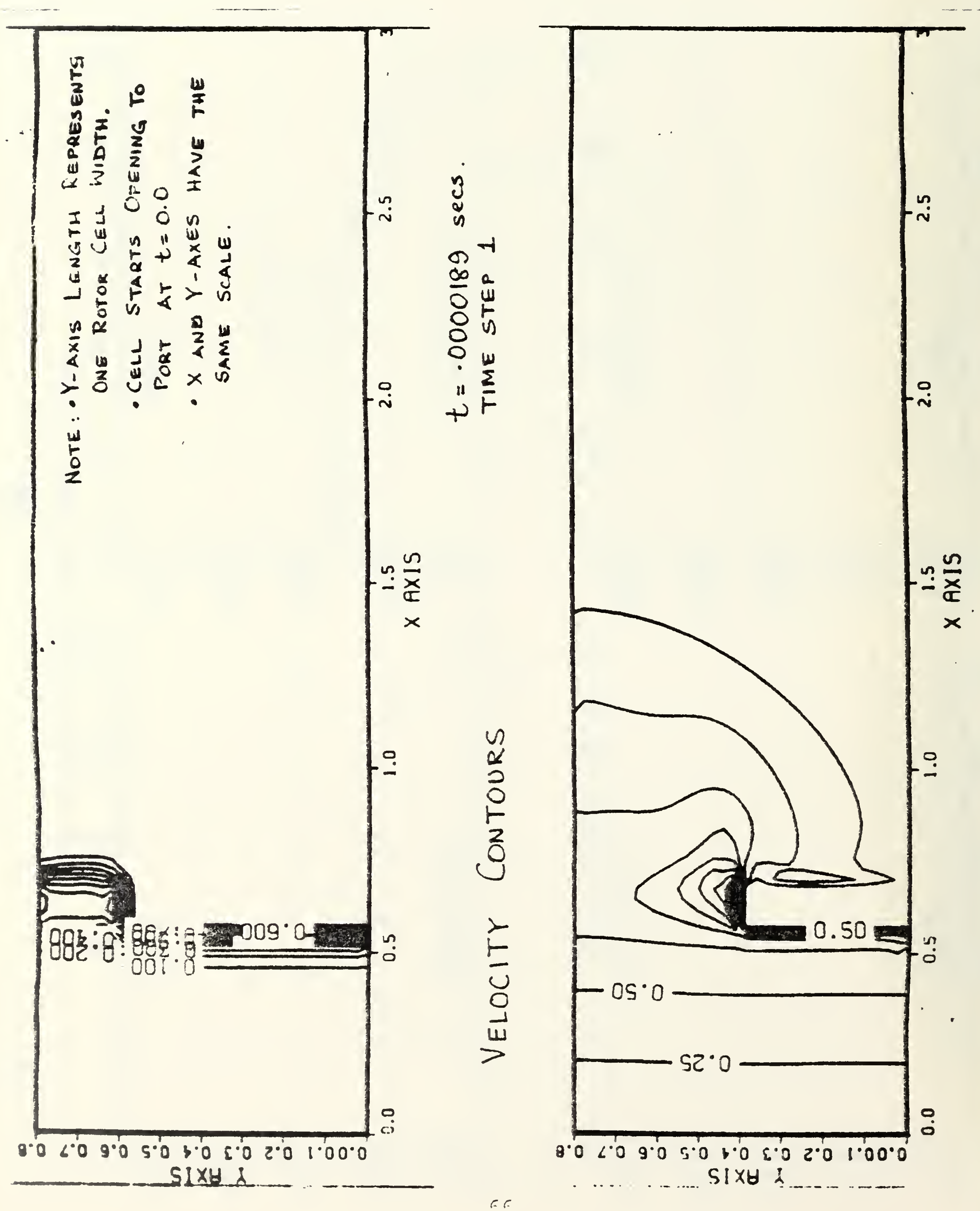


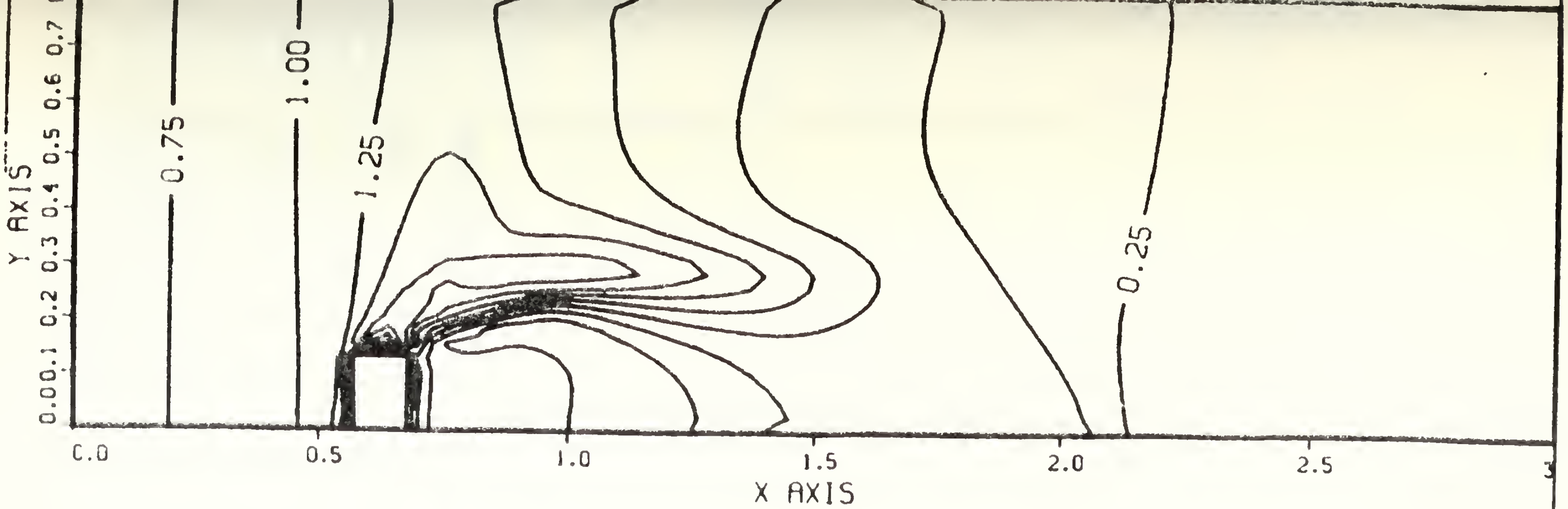


VELOCITY
ENTROPY
DENSITY RATIO
PRESSURE RATIO
P= 2.00
T= 1.25
t1= 32.00
t2= 56.00

APPENDIX E

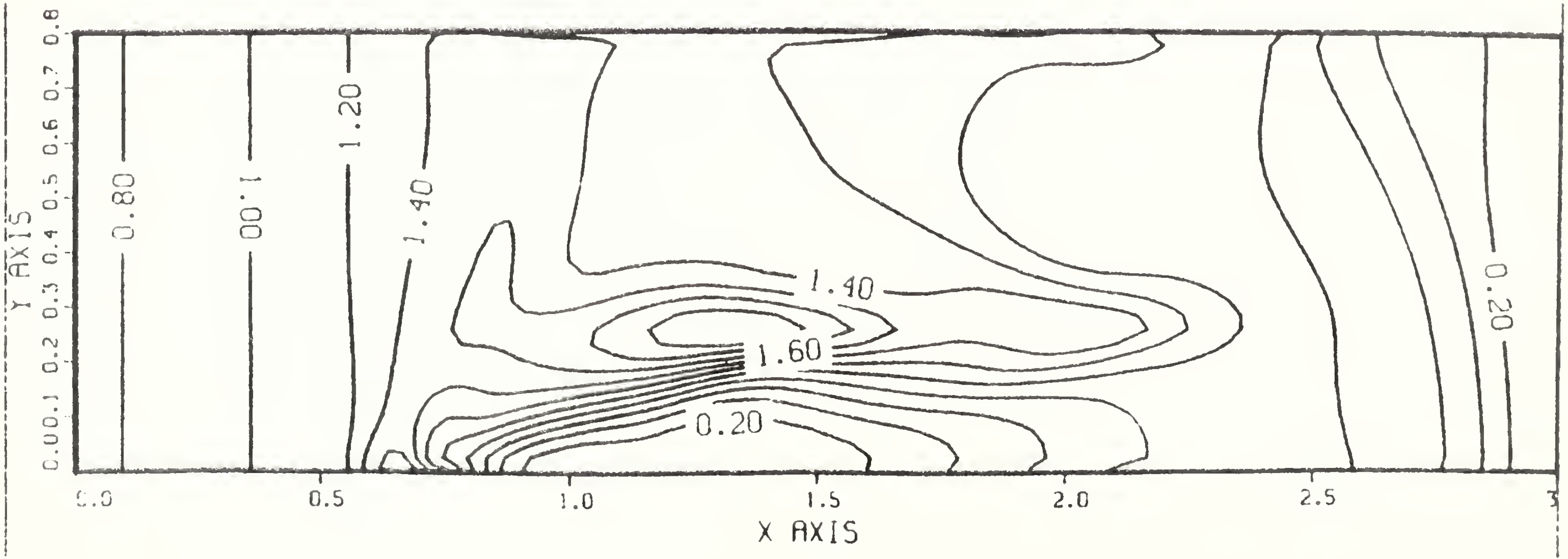
PRELIMINARY COMPUTATIONAL RESULTS
FOR CELL OPENING/CLOSING PROCESS



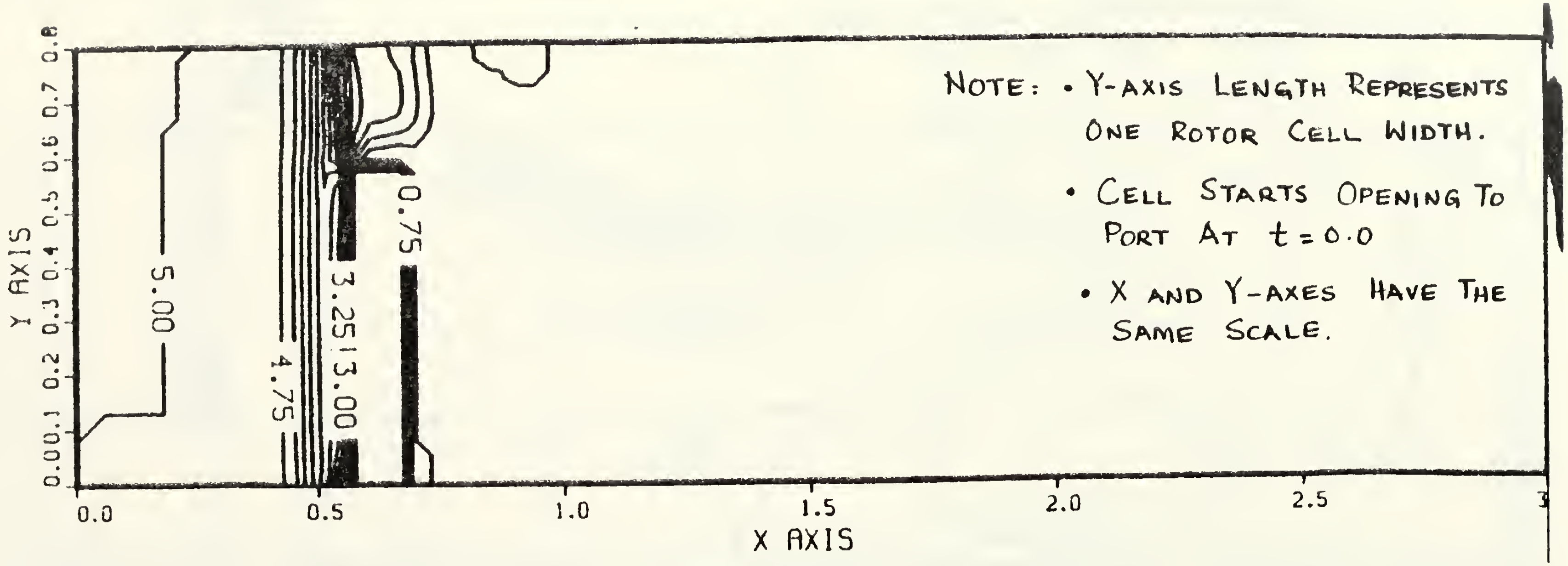



t=.0000539 secs.
TIME STEP 3

VELOCITY CONTOURS

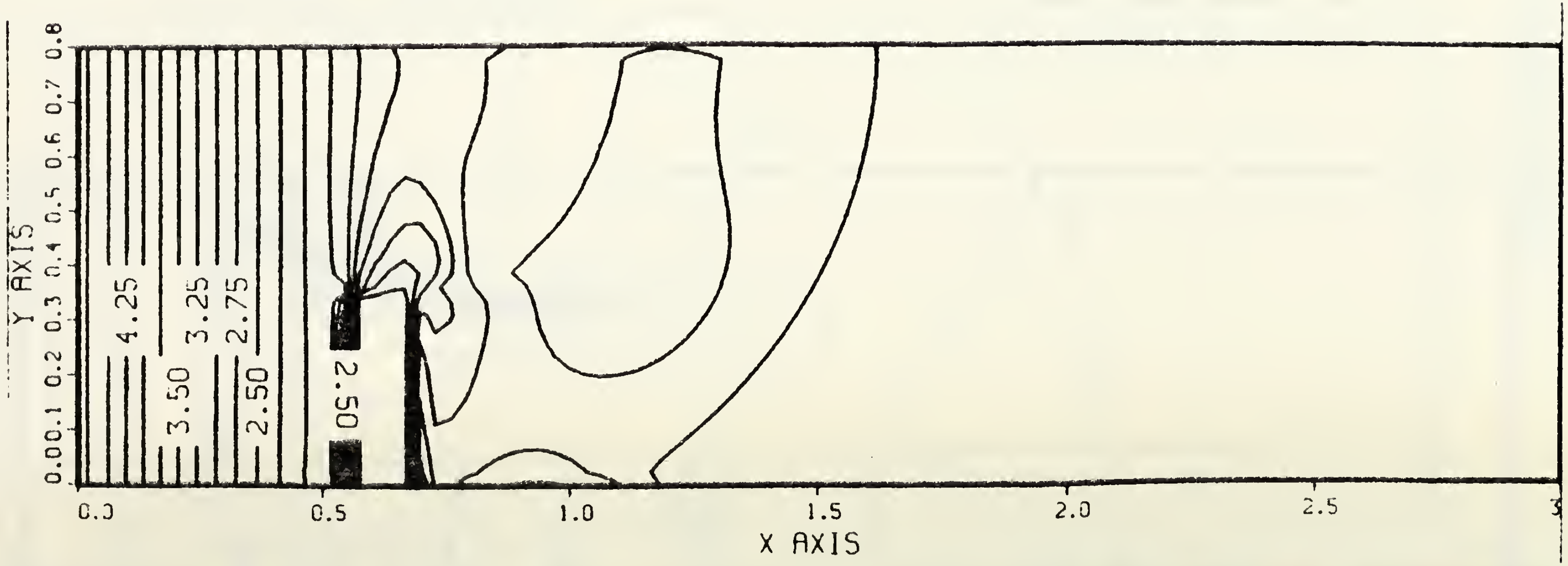



t= .0000711 secs.
TIME STEP 4

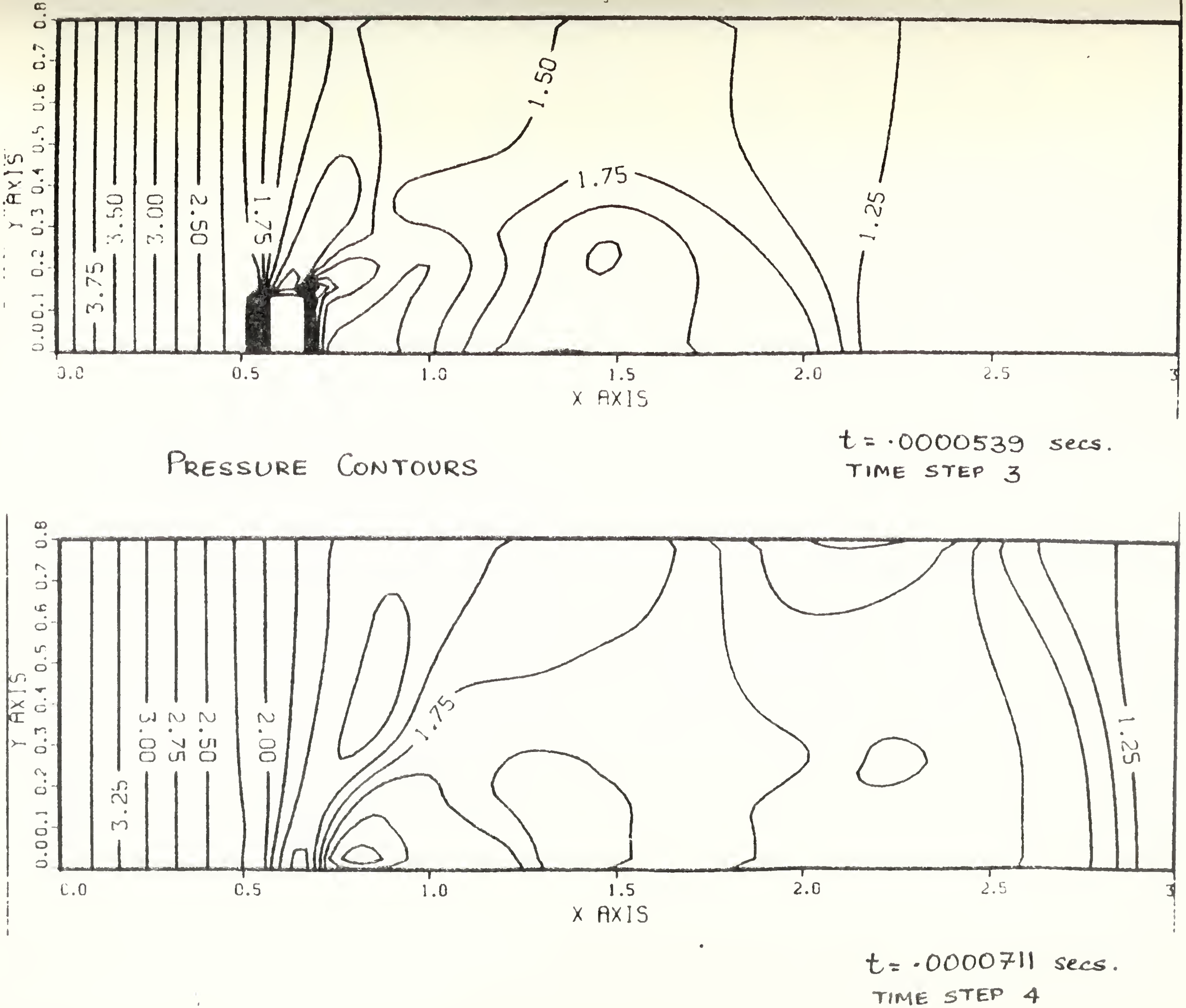



t = .0000189 secs.
TIME STEP 1

PRESSURE CONTOURS




t = .0000382 secs.
TIME STEP 2




Pressure Contours

## APPENDIX F

## THE SIMPLE FOUR-PORT DIRECT-FLOW ROTOR

The simplest arrangement of ports, waves and interfaces which can be used in a direct flow wave rotor is shown in Fig. F1. The process is illustrated on a T-S diagram in Fig. F2 and the velocity diagrams at the ports are illustrated in Fig. F3.

The states shown at stations $Q_a$ and $Q_g$ in Fig. F1 are quiescent relative to the rotor, the relative flow velocities $W_1 = W_4$ and $W_2 = W_3$. From continuity across $Q_a$, the flow rate of air, $\dot{w}_a$, is given by

$$\dot{w}_a = \rho_{qa} \cdot V_c \cdot N_u \qquad F(1)$$

where $\rho_{qa}$ is the density of the air, $V_c$ is the cell volume and $N_u$ is the rotor velocity expressed as cells per unit time. Similarly, continuity across $Q_g$ requires that the flow rate of gas, $\dot{w}_g$, is given by

$$\dot{w}_g = \rho_{qg} \cdot V_c \cdot N_u \qquad F(2)$$

where $\rho_{qg}$ is the density of the quiescent gas in the passages.

From Eq. F(1) and Eq. F(2),

$$\frac{\dot{w}_a}{\dot{w}_g} = \frac{\rho_{qa}}{\rho_{qg}} \qquad F(3)$$

Since the pressures at stations $Q_a$ and $Q_g$ are of comparable magnitudes, differences in temperature from the heat addition will cause the ratio $\frac{\rho_{ga}}{\rho_{gg}}$ to become larger than unity. Equation F(3) then requires that $\dot{w}_a > \dot{w}_g$ so that not all the air which is compressed can be passed back through the rotor.

The rate of work which can be obtained from the shaft can be calculated from momentum considerations. Since the flows at the ports are steady, the axial moment exerted on the shaft in the direction of rotation, $M_a$, is given by:*

$$M_a = (RV_{u_1} - RV_{u_2})\dot{w}_a + (RV_{u_3} - RV_{u_4})\dot{w}_g \qquad F(4)$$

where R is the radial distance, so that if $\omega$ is the rotation rate the rate of work output on the shaft, $\dot{W}_o$ is given by

$$\dot{W}_o = M_a\omega = (UV_{u_1} - UV_{u_2})\dot{w}_a + (UV_{u_3} - UV_{u_4})\dot{w}_g \qquad F(5)$$

In general, from Fig. F3,

$$V_u = W \sin\beta + U \qquad F(6)$$

so that

$$\dot{W}_o = (UW_1 \sin\beta - UW_2 \sin\beta)\dot{w}_a + (UW_3 \sin\beta - UW_4 \sin\beta)\dot{w}_g \qquad F(7)$$

Since $W_4 = W_1$ and $W_3 = W_2$,

$$\dot{W}_o = U\sin\beta(W_1 - W_2)(\dot{w}_a - \dot{w}_g)$$

$$= \dot{w}_a\left[1 - \frac{\dot{w}_g}{\dot{w}_a}\right](W_1 - W_2)U\sin\beta \qquad F(8)$$

It is interesting to note that no work can be obtained if the rotor cells are axial ($\beta = 0$), but that work can be extracted

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*Vavra, M. H., "Aero-Thermodynamics and Flow in Turbomachines," John Wiley & Sons, New York, 1960.

with "straight" passages if they are staggered ($\beta \neq 0$). Equation F(3) shows that as heat is added ($\rho_{qg}$ becoming smaller than $\rho_{qa}$), it is required that $\frac{\dot{w}_g}{\dot{w}_a}$ become progressively smaller than unity. From Eq. F(8), progressively more work can then be extracted on the shaft, with the stipulation that the low pressure scavenging velocity $W_1$ be greater than the high pressure scavenging velocity $W_2$.

For axial passages, when the nett work output is zero from Eq. F(8), the energy from the gas is transferred directly to the air. The steady-flow energy equation then gives

$$\dot{w}_a \, c_{p_a}(T_{t_2} - T_{t_1}) = \dot{w}_g \, c_{p_g}(T_{t_3} - T_{t_4}) \qquad F(9)$$

or

$$\left(\frac{T_{t_3} - T_{t_4}}{T_{t_2} - T_{t_1}}\right) = \frac{c_{p_a}}{c_{p_g}} \cdot \left(\frac{\dot{w}_a}{\dot{w}_g}\right) \qquad F(10)$$

If the efficiencies of compression and expansion are defined in the usual way; referring to Fig. F2,

$$\eta_{c_{t-t}} = \frac{T_{t_{2is}} - T_{t_1}}{T_{t_2} - T_{t_1}} \qquad F(11)$$

is the total-to-total efficiency of compression and

$$\eta_{T_{t-s}} = \frac{T_{t_3} - T_{t_4}}{T_{t_3} - T_{4_{is}}} \qquad F(12)$$

is the total-to-static efficiency of compression.

Using Eq. F(11) and Eq. F(12), Eq. F(10) can be rewritten as

$$\eta_c \cdot \eta_T \cdot \frac{T_{t3}\left(1 - \frac{T_{4is}}{T_{t3}}\right)}{T_{t1}\left(\frac{T_{t2is}}{T_{t1}} - 1\right)} = \left(\frac{c_{p_a}}{c_{p_g}}\right) \cdot \frac{\dot{w}_a}{\dot{w}_g} \qquad F(13)$$

If it is assumed that $p_{t3} = p_{t2}$ and that $p_4 = p_{t1}$, and the total pressure ratio of the cycle is defined as

$$\pi = \frac{p_{t2}}{p_{t1}} = \frac{p_{t3}}{p_4} \qquad F(14)$$

then Eq. F(13) can be written as

$$\frac{T_{t3}}{T_{t1}} = \left(\frac{\pi^{\frac{\gamma-1}{\gamma}}}{\eta_{c_{t-t}} \cdot \eta_{T_{t-s}}}\right) \cdot \left(\frac{c_{p_a}}{c_{p_g}}\right) \cdot \left(\frac{\dot{w}_a}{\dot{w}_g}\right) \qquad F(15)$$

Since the air to gas weight flow ratio is controlled by the heat added through Eq. F(3) the expression in Eq. F(15) relates the heat which can be added in such a cycle to the pressure ratio and energy transfer efficiencies. However, it is possible that in practice, Eq. F(15) can not be satisfied for pressure ratios and efficiencies which are realistic.

The particular case of a 4-port rotor with axial passages has been considered in an approximate way by Hussmann (Ref. 19) and more recently by Berchtold (Ref. 17). Hussmann has also discussed, somewhat empirically, the apparent inability of the simple 4-port machine to maintain proper scavenging without recourse to external aids such as blowers or fans. It is possible that for effective operation with good scavenging, both outgoing velocities (and especially, the low pressure scavenge velocity) may have to be increased.

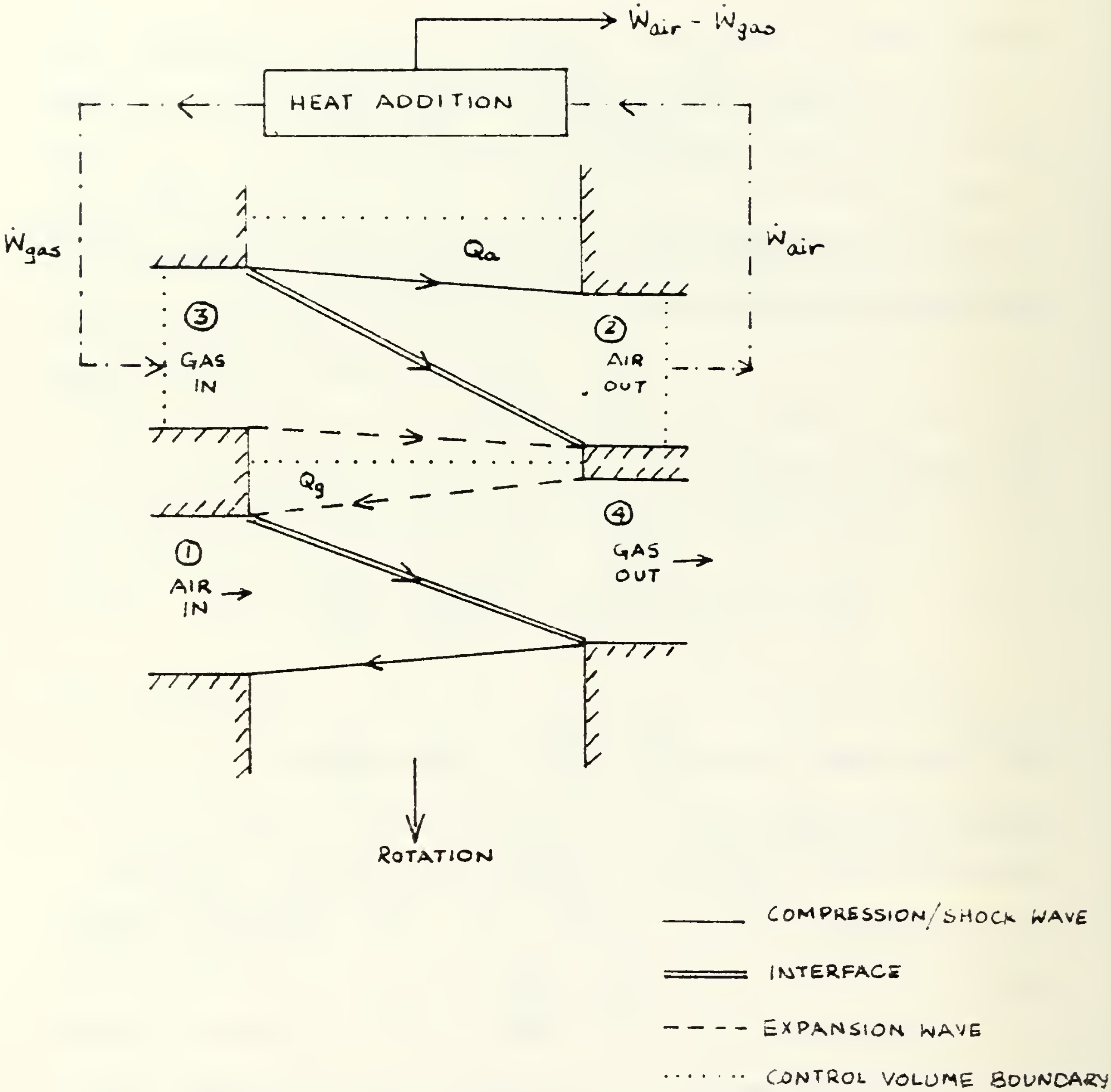



FIG F1 : SIMPLE 4-PORT DIRECT-FLOW MACHINE

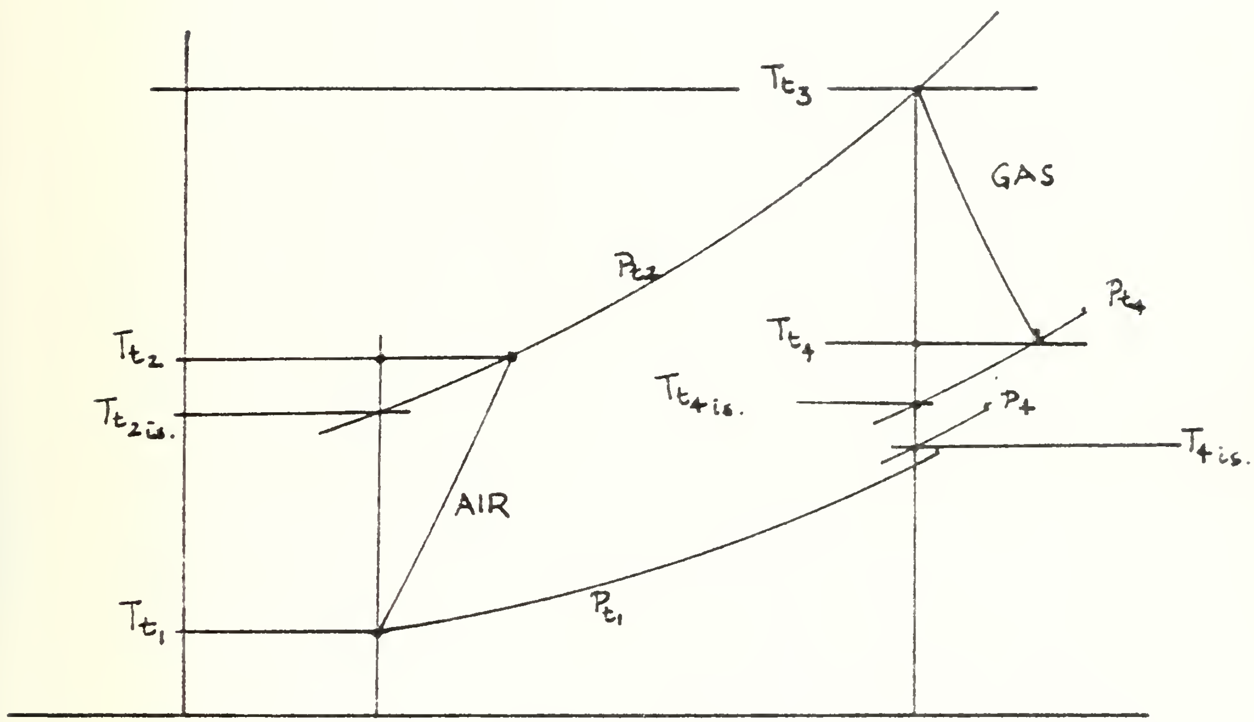
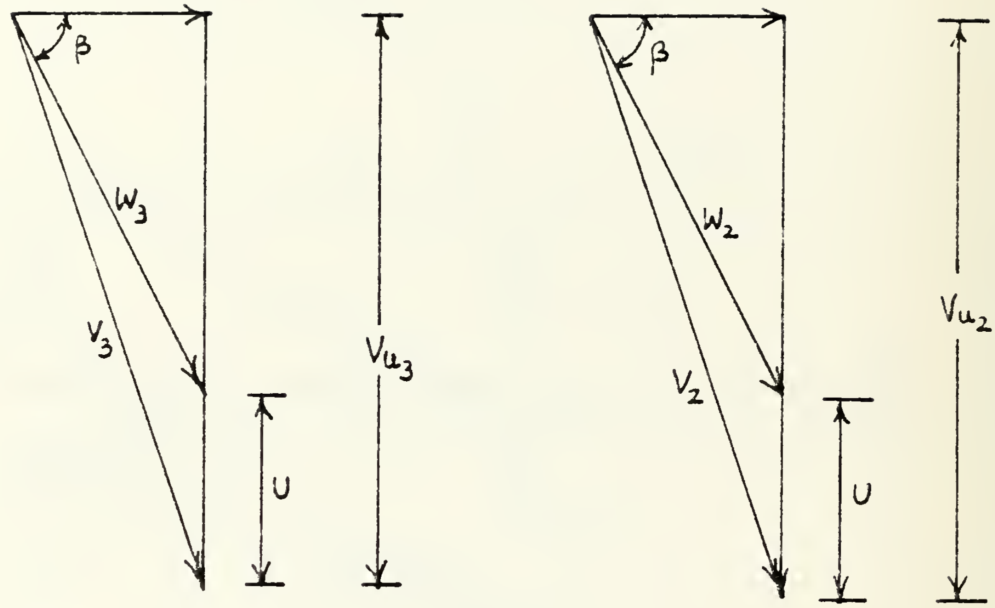
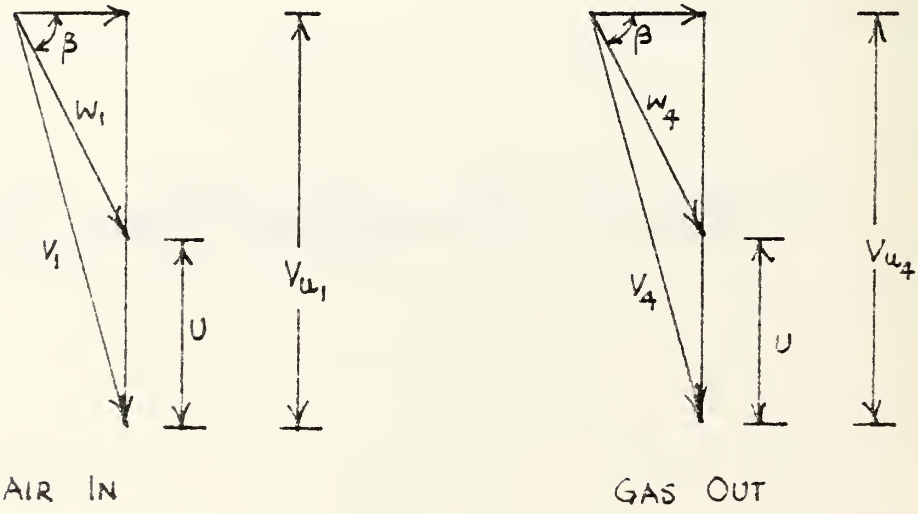



FIG. F2 : TEMPERATURE-ENTROPY DIAGRAM




GAS IN AIR OUT

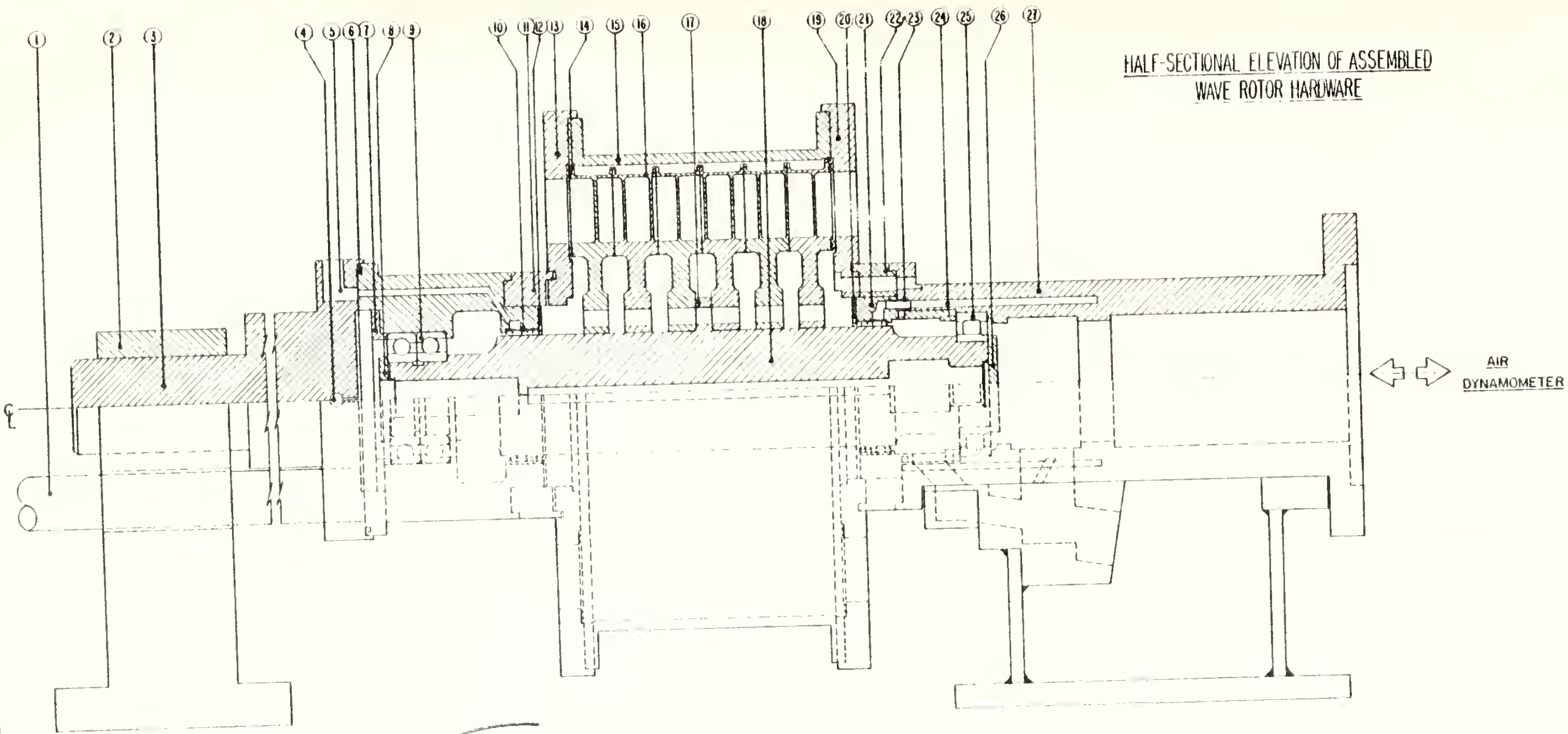
FIG. F3 : VELOCITY DIAGRAMS AT THE PORTS

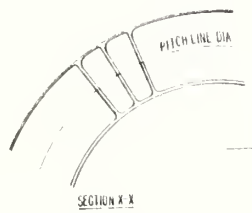


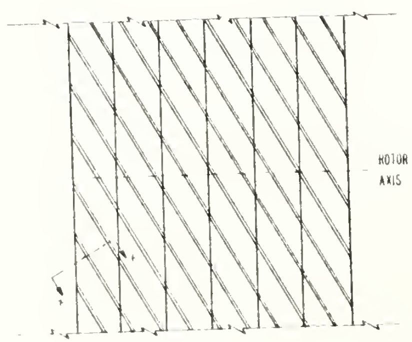

AIR IN GAS OUT

APPENDIX G

HALF-SECTIONAL ELEVATION OF ASSEMBLED WAVE ROTOR HARDWARE




PARTIAL VIEW OF ROTOR DEVELOPMENT AT PITCH LINE DIAMETER

NOTE ASSEMBLY OF SIX WAFERS TO FORM STRAIGHT PASSAGES STAGGERED AT 60° TO ROTOR AXIS

| SELECTED DIMENSIONS | |
|---|---|
| ROTOR PITCH LINE DIAMETER | 7.0 in. |
| ROTOR WIDTH (ALL 6 WAFERS) | 4.8 in. |
| ROTOR PASSAGE LENGTH | ≈11.0 in. |
| ROTOR PASSAGE HEIGHT | 1.2 in. |
| ROTOR PASSAGE WIDTH | ≈0.365 in. |
| NUMBER OF PASSAGES | 24 |

| | |
|---|---|
| 27 | ROLLER BEARING HOUSING |
| 26 | BEARING RETAINER PLATE |
| 25 | ROLLER BEARING |
| 24 | OIL HOLE |
| 23 | HOLLOW PIN |
| 22 | ADAPTER RING |
| 21 | LABYRINTH HOUSING |
| 20 | LABYRINTH RETAINING PLATE |
| 19 | STATOR PORT PLATE |
| 18 | ROTOR SHAFT |
| 17 | ROTOR LOCATING FLANGE |
| 16 | ROTOR (6 WAFERS) |
| 15 | ROTOR HOUSING |
| 14 | STAGGERED LABYRINTH SEAL |
| 13 | STATOR PORT PLATE |
| 12 | LABYRINTH RETAINING PLATE |
| 11 | ADAPTER RING |
| 10 | STRAIGHT-THROUGH LABYRINTH |
| 9 | DUPLEX BALL BEARINGS |
| 8 | BEARING CAP |
| 7 | BEARING RETAINER PLATE |
| 6 | O-RING |
| 5 | COOLING AIR INTAKE |
| 4 | OIL INTAKE |
| 3 | SUPPORT BLOCK |
| 2 | SUPPORT BLOCK |
| 1 | DRAIN LINE |

PARTS LIST

ASSEMBLY: WAVE ROTOR HARDWARE

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